AIRCRAFT HYDRAULICS

BY

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PREFACE

Aircraft have increasing numbers of services that require remotely controlled power operation. A large percentage of these services can best be performed hydraulically.

Believing that the average American engineer, mechanic, or student wants to know, not simply the details of systems in current use and arbitrary rules for their design, but also the basic principles and general rules for their design, and that thus equipped he can understand present systems and develop future systems, the author in this book has concentrated on the principles underlying the design of aircraft hydraulic systems rather than on the details of systems in current use.

The author wishes to acknowledge his deep indebtedness to Arthur E. Raymond, whose recognition of the merits of hydraulic systems has kept the Douglas Aircraft Company leading in advances in the application of hydraulic systems in aircraft.

HAROLD W. ADAMS.

Santa Monica, Calif.
January, 1943.

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CHAPTER I

INTRODUCTION

The aircraft hydraulic systems that this volume will cover can be defined as systems for the transmission of power by means of fluid under pressure. To fulfill the requirements of this definition, the system must consist of at least three parts, as follows: a pump, a transmission line, and a motor. It must include a pump because the pump is the unit that puts work into the fluid, *i.e.*, takes mechanical power from whatever source it comes and uses it to move fluid under pressure. It must include a line, or pipe, for transmitting the fluid under pressure from one place to another, and it must include a motor for transforming the moving fluid under pressure back into mechanical power.

Thus, in an aircraft hydraulic system, which is a system for transmitting power by means of fluid pressure, it is necessary to transform the power into fluid pressure, move this a certain distance, and then transform it back into power again.

The advantages of such a system of power transmission are many; there also are limitations, which must be recognized. In deciding whether to use a hydraulic system rather than some other form of power-transmission system, such as mechanical or electrical, the relative advantages and disadvantages of each system must be weighed one against the other. Therefore, the first task of a book about airplane hydraulic systems must be to show that such systems have enough advantages to make them the best form of power-transmission system for certain applications.

The chief advantages of the hydraulic form of power transmission are light weight, controllability, and low inertia of the moving parts.

The advantage of light weight in airplanes is obvious. However, the weight advantage of hydraulic systems is confined to the high powers. In the lower powers, hydraulic systems, because of the minimum sizes of the parts available or practicable, become increasingly heavy when compared with competing systems of power transmission.

A very high degree of controllability characterizes hydraulic systems and makes them the logical answer to such airplane problems as power flight controls, power gun turrets, and power brakes.

The advantage of having moving parts of low inertia is of importance when it is necessary to bring some mechanism, such as a retractable landing gear, to a stop in an exact position. Under such conditions, light parts moving at low speed, such as a piston in a cylinder, can be stopped by a simple mechanical stop at the end of their travel, whereas a high-speed rotating part, having considerable inertia, must be brought to a stop gradually or disruptive forces will result.

Other advantages and limitations will present themselves as specific cases are studied and when hydraulic systems are compared with other competing systems of power transmission.

An example of the simplest form of such a system is the hydraulic-brake system in an airplane or automobile in which the pump, which is usually called the *master cylinder*, is operated by the driver's foot. It forces oil through lines to the motor, in this case called the *brake cylinder*, located at the wheel, where the moving fluid is transformed back into work and applies the brake shoes.

Hydraulic systems are commonly used in aircraft for transmitting power from a pump mounted on one or more of the airplane's main engines to other parts of the airplane, to retract landing gears, operate wing flaps, move gun turrets, and in general wherever high power or a high degree of controllability is required. Other power sources than the airplane's main engines are sometimes used, such as auxiliary engines or electric motors.

In the basic airplane hydraulic system, as diagramed in Fig. 1, there are two additional elements, which are present in most hydraulic systems. These are the reservoir and the directional control valve. The reservoir is necessary in most systems to take care of the change in volume of the fluid with change in temperature. It also takes care of the change in volume of the system that occurs when the operating cylinder is moved and acts to supply fluid to make up losses due to leakage and evaporation.

A directional control valve, or "four-way" valve, is required in any system that must operate in two directions unless a reversible pump is used.

In the diagram of a basic aircraft hydraulic system shown in Fig. 1, *A* is the reservoir, *B* is the pump, *C* is the operating cylinder, *D* is the control valve, and *E*, *E* are lines.

Arrows show the direction of flow of the fluid, which always flows in one direction above the directional control valve but in both directions beyond the valve, depending on valve position. The part of the system above the valve, in which the fluid flows in only one direction, is called the *power system*. The part beyond the directional control valve is known by the name of the unit that it operates, as the *landing-gear system* or the *flap system*.

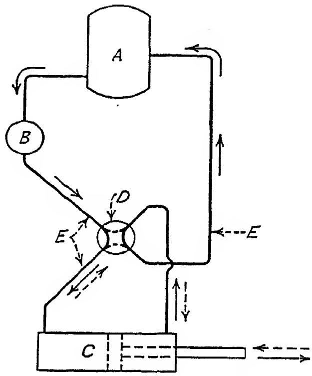


FIG. 1.—Schematic diagram of basic airplane hydraulic system.

The operation of the system may readily be understood by studying Fig. 1. When the control valve is in the position shown by the solid lines, fluid drawn from the reservoir by the pump is forced through the valve into the left-hand end of the operating cylinder, forcing the piston to the right as shown by the solid arrow. The fluid from the right-hand end of the cylinder is returned to the reservoir through the control valve, as shown by the solid arrows. When the control valve is moved to the position shown by the dotted lines, fluid drawn from the reservoir by the pump is forced through the valve into the right-hand end of the operating cylinder, forcing the piston to the left as shown by the dotted arrow. The fluid from the left-hand end of the cylinder now returns to the reservoir through the control valve, as shown by the dotted arrows. This basic system is the one from which any hydraulic system can be derived. Additions may be made to it for the purpose of providing additional sources of power, operating additional cylinders or motors, making operation more automatic, or increasing the reliability; but these additions are all made on the framework of the basic hydraulic system diagramed in Fig. 1.

The following brief description of the units may help those not familiar with modern aircraft hydraulic systems to visualize the system as it might be installed in an airplane. The reservoir

usually consists of a welded aluminum tank. The most desirable shape is a cylinder with its axis vertical. The tank in the Douglas DC-3 airplane, for example, is about 7 in. in diameter and 20 in. in height.

The reservoir tank must have a filler opening where oil can be added to the system. It must have a drain for changing the oil in the system and for draining out the sediment that collects in the bottom of the tank and a vent line to open the tank to the atmosphere to prevent changes in pressure caused by changes in the internal volume. It has one or more outlets to the pumps and one or more returns from the directional control valves. It usually has some sort of level-measuring device, such as a gauge glass or a calibrated stick, and it sometimes, though not always, includes some sort of strainer. A typical reservoir is shown in Fig. 2.

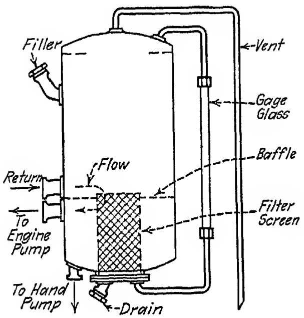


FIG. 2.—Schematic drawing of typical hydraulic-system reservoir.

Hydraulic-system pumps may be either power-driven or hand-operated pumps or, in the case of brake systems, pedal-operated pumps.

Hand pumps nearly always take the form of a piston reciprocating in a cylinder provided with inlet and outlet check valves. A check valve is a device that allows fluid to travel freely in one direction but stops its flow in the other direction. A typical hand pump is shown in Fig. 3. This pump has a displacement of $\frac{3}{4}$ cu. in. per stroke and weighs 3 lb.

Engine pumps may be of three general types. A very few high-pressure small-displacement pumps are built with reciprocating pistons and check valves. A somewhat larger number are built with reciprocating pistons and rotary valves. Some sort of valving is, of course, necessary to prevent the fluid in the pressure line from reentering the cylinder on the intake stroke. The great majority of aircraft hydraulic-system engine pumps, however, are gear pumps, in which oil trapped in the teeth of gears is forced

from the low-pressure to the high-pressure side of the pump. A typical gear pump is shown in Fig. 4. This pump, which is about average in size, has a displacement of 5 gpm. at 3000 rpm. and will deliver this flow at up to 1000 lb./sq. in. pressure. The horsepower

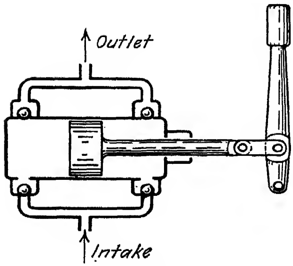


FIG. 3.—Schematic drawing of typical hand pump.

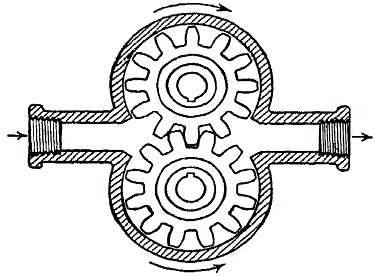


FIG. 4.—Schematic drawing of typical gear pump.

required under these conditions is about 5, which, since the pump weighs 5 lb., makes a weight/horsepower ratio of 1.

In aircraft hydraulic systems the motor for transforming moving fluid under pressure back into work usually takes the form of a cylinder with a piston moving in it; for the most common applications for hydraulic systems are in the operating of retractable landing gears, wing flaps, bomb doors, and similar services in which the unit to be operated has to be moved through a certain travel and back again. The piston in these operating cylinders is generally packed with some sort of nonmetallic packing to prevent fluid leakage. Cylinder dimensions are usually given as bore and stroke, which vary from 1 by 2 in. up to 5 by 30 in., depending on

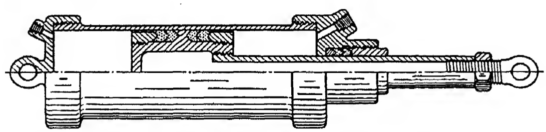


FIG. 5.—Schematic drawing of typical operating cylinder.

the power required. Most operating cylinders are hinged or pivoted at the ends, and fluid is conveyed to them through flexible hoses. A typical operating cylinder is shown in Fig. 5.

Sometimes, where rotary motion is desired, hydraulic motors are used, which are simply gear or piston pumps with oil introduced

into the pressure port so that the fluid backing up against the pistons or gears makes them operate.

A directional control valve is necessary wherever the direction of fluid in lines must be reversed, as when an operating cylinder must be operated positively in both directions. Directional control valves have taken various forms. Some years ago, plug valves were extensively used. In the plug valve, the body has a tapered hole that is intersected by four holes at right angles to it. A plug fits the tapered hole and has two cutouts in its sides. When it is moved in one direction, it connects the pressure port with one cylinder port and connects the other cylinder port with the reservoir port. When it is turned 90 deg , it reverses the cylinder connections with respect to the pressure and reservoir ports (see Fig. 6).

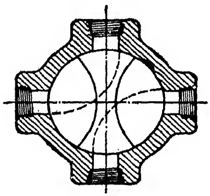


Fig. 6.—Schematic drawing of typical plug valve.

The valves in most common use in modern systems are slide valves and poppet valves. A slide valve consists of a bored hole having a close-fitting piston. The piston has lands which fit tightly, and grooves or recesses through which the oil can flow. The various ports are drilled into the bore. The piston in various positions thus connects various ports through its grooves while the remaining ports are sealed off by the tight-fitting lands on the piston. A valve of this type is shown in Fig. 7.

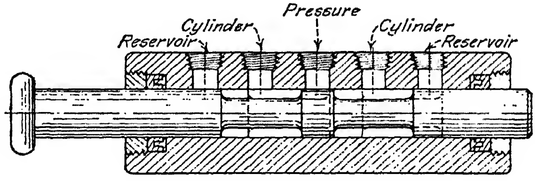


Fig. 7.—Schematic drawing of typical slide valve.

Poppet valves use a poppet, or ball, moving on or off seats to control the flow between ports. This can best be seen in the diagram of a poppet valve in various positions in Fig. 8.

Poppet valves vary in size from about 2 to 6 in. in length and from 1 to 5 lb. in weight and have port openings from 1/8 in. diameter to ½ in. or larger.

The fluid is conveyed between the various units by lines, or pipes, consisting of tubing, usually of some soft material such as 52SO aluminum alloy, that varies in size from ¼ in. outside

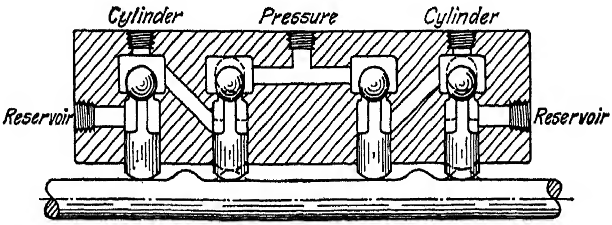


FIG. 8.—Schematic drawing of in-line type poppet valve.

diameter on operating lines on small airplanes to 1½-in. lines for pump suction lines on large airplanes. Most pressure lines are in the range from 5⁄16 to ½ in. outside diameter, and suction lines range from ½ to 1 in. outside diameter. The tubing is connected to other tubing or to units by flared fittings in which a flare on the end of the tubing acts as both a mechanical connection and a gasket to seal against leakage. This flare is brought up against the fitting by a nut. In some designs a sleeve is interposed between the nut and the back of the tube flare. A typical tube fitting is shown in Fig. 9.

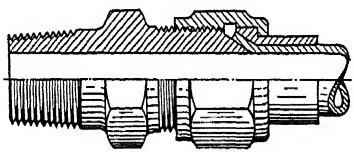


FIG. 9.—Sectional drawing of typical tube fitting.

The fluid used in the system is usually a light mineral oil, somewhat thinner than SAE 10 automobile-engine lubricating oil. It was originally used in Sperry automatic pilots and so is often referred to as "Sperry oil." Its properties will be discussed in detail later.

CHAPTER II

FLOW OF FLUIDS

A knowledge of the principles governing the flow of fluids is necessary for an understanding of the functioning and design of hydraulic systems. The flow of fluids has been studied since the days of the Romans, but the science of hydrodynamics did not begin to assume its present form until the time of Osborne Reynolds, whose studies laid the foundation upon which the work of later investigators has been based. Only recently, aided by the intensive work done by such men as Prandtl and von Kármán in the related field of aerodynamics, the science of hydrodynamics has taken another great stride and has at last assumed the proportions of an exact science, in which experiment can be correlated with theory.

Hydrodynamics can be classified in two major divisions, flow in pipes and flow under local disturbances, such as change of section in pipes and through orifices. Both these divisions are of importance in airplane hydraulic systems, and both must be fully understood before attempting to design an airplane hydraulic system, unless cut-and-try methods are to be used.

FLOW IN PIPES

The study of flow in pipes is of major importance in aircraft hydraulics, for most units to be operated are at some distance from the source of power, and the intervening distance must be bridged by lines, through which oil flows when the system is in operation. Because resistance is created whenever there is a flow of fluid in a pipe, it is essential that the amount of this resistance be calculated in order to determine how much of the power available will be absorbed in overcoming resistance and how much will remain to do useful work.

Resistance to flow in pipes is caused by the shearing action that takes place in the flowing fluid and between the fluid and the wall of the tube. Because a stationary body of fluid has no strength in

shear, it follows that resistance to flow in pipes is always dynamic. Without motion there can be no resistance. In a pipe in which the fluid is stationary the pressure is the same throughout the fluid. This is one of the fundamental principles of hydrostatics, the study of fluids at rest. In hydrodynamics, the study of fluids in motion,

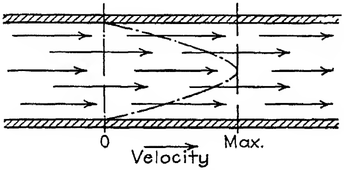


FIG. 10.—Conditions in laminar flow.

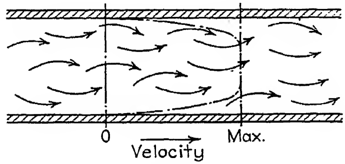


FIG. 11.—Conditions in turbulent flow.

this is of no interest, except to indicate that when fluids are flowing at extremely low velocities the probability is that the resistances will be negligible.

The flow of a fluid in a pipe may be of two different types, laminar and turbulent, there being an abrupt transition between types. As fluid starts into motion from a state of rest, the flow is uniform, or streamline, in character. All particles move in parallel streams, and the shearing is equal between adjacent streams. The velocity increases uniformly toward the center of the stream, as shown in Fig. 10.

This condition is maintained as long as the flow continues slow. As the velocity of flow is increased beyond a certain critical speed, mixing, or turbulence, starts to take place in the stream and the particles of fluid no longer move in parallel lines but start to move back and forth across the pipe as well. This results in a more nearly uniform velocity across the stream, as shown in Fig. 11.

Consequently, close to the wall of the pipe there is greater shearing velocity, and therefore increased resistance to flow, for the resistance to shearing increases with velocity. The characteristic of a fluid that determines its resistance to shear is its viscosity.

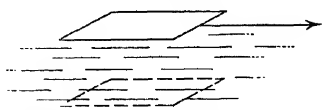


FIG. 12.—Measurement of fluid viscosity.

If there are two plates of a given size with fluid between them, as in Fig. 12, and one of the plates is moved at a constant velocity, a certain amount of force is required, caused by the resistance to shear of the fluid between the plates.

A fluid of unit viscosity is one that requires a unit force to move plates of a unit size at a unit speed through the fluid. If the units taken are English units, the plates will be 1 ft. square, 1 ft. apart, and will move at a velocity of 1 ft./sec. when a force of 1 lb. is applied, and the unit of viscosity will be a viscosity in lb.-ft. sec., sometimes referred to as a viscosity of "one engineering unit." If the fluid has twice the resistance to shear of a fluid of unit viscosity, a force of 2 lb. would be required to move the 1-ft.-square plates at 1 ft./sec. and the viscosity will be two engineering units.

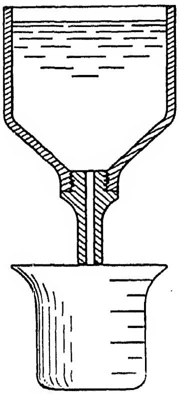


FIG. 13.—Schematic drawing of Saybolt viscosimeter.

If the units are taken in c.g.s. (centimeter-gram-second) units, the plates will be 1 cm. square, 1 cm. apart, and will move at a velocity of 1 cm./sec. when a force of 1 dyne is applied. (One dyne is the unit of force in the c.g.s. system and equals $\frac{1}{980}$ gram.) This unit of viscosity in the c.g.s. system has been given the name of *poise*, after Poiseuille, who did much of the early work on fluid resistance. A unit of more convenient size that is often used is the *centipoise*, which is 0.01 poise.

The viscosity of a fluid is never determined by moving plates but is determined by measuring the flow through a pipe and converting the results into viscosity. If a given pressure were used to force fluid through a pipe, the rate of flow would be proportional to viscosity. However, it is more convenient to construct a measuring device using a certain head, or height, of fluid, and most viscosity-measuring devices are constructed on this principle, as, for example, the Saybolt viscosimeter shown in Fig. 13.

This introduces another factor, *density of the fluid*, into the measurement of viscosity, for a constant *height* of fluid is used. If a given volume of one fluid weighs more than the same volume of another fluid, obviously the heavier fluid will run out through the tube faster, because there is more pressure forcing it through. To eliminate this factor of density the viscosity obtained from the viscosimeter—which for the heavy fluid appeared lower than it really was because the fluid flowed out faster, just as it would if the fluid has been a lighter fluid of lower viscosity—must then be multiplied by the ratio of the density of the heavy fluid to the

density of the light fluid, or, to put it in general terms, by the ratio of the density of the tested fluid to the density of a standard fluid, such as water.

The viscosity, as measured by the viscosimeter and thus including the effect of density, is called the *kinematic viscosity*, to distinguish it from the *true*, or *absolute, viscosity*. The unit of kinematic viscosity is the absolute viscosity divided by the density.

$$\nu = \frac{\mu}{\rho}$$

where ν = kinematic viscosity
μ = absolute viscosity
ρ = density
all in any consistent system of units.

In English units the kinematic viscosity is expressed as the absolute viscosity in lb. sec./ft.^{2} divided by the density in slugs per cubic foot. [A slug is the unit of force divided by the acceleration of gravity (32.2 ft./sec.^{2}) and equals 32.2 lb.] The kinematic viscosity is in units of ft.^{2}/sec.

The characteristic of fluid flow that determines whether the flow is turbulent or laminar is the ratio of inertial forces to viscous forces. When the viscous forces predominate, the flow is smooth, or laminar; when the inertial, or weight and velocity, forces predominate, the flow becomes turbulent. This ratio is called *Reynolds number* after Osborne Reynolds, who first discovered the importance of this ratio in classifying types of fluid flow. It is designated by the letter R and is written as

$$R \quad \frac{Vd\rho}{\mu} = \frac{\text{velocity} \times \text{diameter} \times \text{density}}{\text{absolute viscosity}}$$

Since ν, the kinematic viscosity, $= \mu/\rho$, it can be substituted in the formula, which then takes the form

$$R = \frac{Vd}{\nu} = \frac{\text{velocity} \times \text{diameter}}{\text{kinematic viscosity}}$$

This ratio is nondimensional; *i.e.*, when a consistent system of units is used, the result R is the same.

The general formula for resistance to flow through a pipe is

$$h = f\,\frac{V^2L}{d2g}$$

where V = velocity of flow.
L = length.
d = diameter of pipe.
g = acceleration of gravity.
f = friction or resistance factor (experimentally determined).
h = resistance to flow through a pipe expressed as head of fluid lost in a length of pipe L.

This can be shown graphically by imagining a pipe with two standpipes on it, spread a distance L apart. The difference in height to

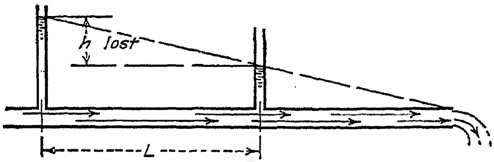


FIG. 14.—Head loss due to fluid friction.

which the fluid will rise in the pipes represents the head lost owing to resistance or fluid friction in the pipe, as in Fig. 14.

Since it has been seen that the type of flow depends on the Reynolds number, it should be possible to plot f against R and get consistent results. This has been done by numerous experimenters, with consistent results, which produce the curve shown in Fig. 15.

It can be seen from this curve that there is a very definite change from laminar to turbulent flow in the region of $R = 2000$. It can also be seen that near the critical region the flow is unstable and may be either laminar or turbulent. To be conservative in this region and to simplify the calculations for resistance the turbulent-flow line is often extended into the region of lower Reynolds numbers until it intersects the laminar-flow line as shown by the dot-dash line in Fig. 15. This gives a Reynolds number for the change from laminar to turbulent flow of 1080.

In aircraft work, it is usually unnecessary to consider the effect of roughness inside the pipe on resistance to flow. The effect of

roughness is small in the laminar-flow region but may increase the value of f up to double in the turbulent region in the case of very rough pipes.

Formulas have been derived from experimental results and theory for variation of f with R for smooth pipes.

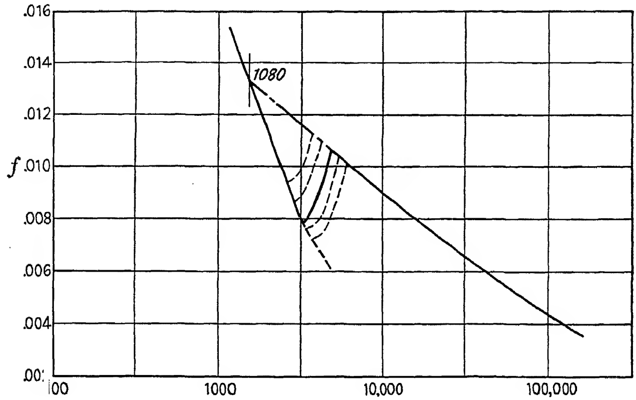


FIG. 15.—Friction factor *vs.* Reynolds number. Refer to formulas for exact values.

For laminar flow the formula is

$$f = \frac{64}{R}$$

For turbulent flow the formula is

$$f = 0.0056 + \frac{500}{R^{0.32}}$$

Formulas have also been derived for fluid resistance directly, such as Poiseuille's formula for head loss in the laminar-flow region, and the Bazin, Chézy, and numerous other empirical formulas for fluid resistance in the turbulent-flow region.

Because for rapid calculation of fluid resistance it is desirable to construct a chart from which the head loss or, as is more convenient in aircraft work, the pressure loss, can be read directly, these formulas are of little interest here. Therefore, only the formulas previously given, which cover all cases and consequently

can be used when the problem in hand is outside the range of the chart, will be considered, in order to avoid confusion.

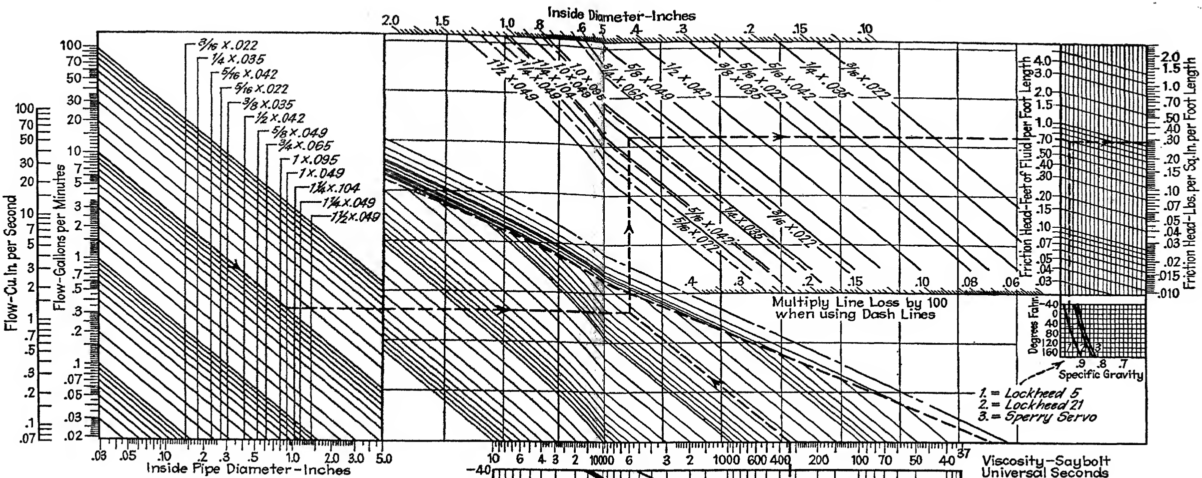
Such a chart, which can be used for rapid determination of pressure loss due to flow in pipes, is given in Fig. 16. The formulas given above must be used when a problem is outside the range of the chart.

The viscosity is given on this chart in S.S.U., or Saybolt universal seconds, which is the unit of kinematic viscosity most commonly used in the United States; it is also given in centistokes (c.g.s.) and in English units.

The chart is used as follows: Suppose it is desired to find the head or pressure lost when a light oil (Sperry Servo oil) at a temperature of 70° flows through a 1-in. tube 15 ft. long at a rate of 4 g.p.m. Starting at 4 g.p.m. on the left-hand scale of the chart, travel down and toward the right along the sloping line until the vertical line at 1 in. inside diameter is intersected. From this point, travel to the right until the line sloping up and left from 170 S.S.U. (the viscosity of Sperry Servo Oil at 70°) is intersected. From this point, travel vertically upward until the line running downward and to the right from 1 in. inside diameter is intersected. (In some cases the dotted line running up and left from the lower line size scale is intersected first.) From this point, run to the right until the head-loss scale is reached. From this scale the head loss for a 1-ft. length of pipe is read. Multiply this value by 15 (the length of pipe in this case) to get the head loss in 15 ft. of pipe (if the dotted pipe-size diagonal line was used, multiply the head loss by 100).

To get pressure drop instead of head loss, proceed exactly as previously described; but when the head-loss scale is reached continue to run to the right but slightly downward, along the slightly sloping line until the vertical line through the specific gravity of the fluid is intersected, and from this point run horizontally to the pressure-drop scale. From this scale the pressure drop for a 1-ft. length of pipe is read, which must be multiplied by 15 to get the pressure drop for this case (if the dotted pipe-size diagonal line was used, first multiply the pressure drop by 100).

If the chart did not cover the pipe-size, flow, etc., for which it was required to find the pressure drop, it would be necessary to calculate it. To do this, the formulas previously given can be used.



Example

Determine loss in pressure with Lockheed 5 Fluid at 50°F. at a flow of ten gal. per min. thru a 1" O.D. X .049 line.

Solution

Start from the 10 gal. per min. position on left and from the 50°F. on viscosity grid; follow the dash lines in the directions of the arrows and read friction head of .72 ft. per foot run; follow the slant lines to specific gravity at 50°F. and the horizontal lines to .290 lbs. per sq. in. per foot run.

To Use Another Fluid

Plot viscosity of new fluid on temperature-viscosity grid; do not extrapolate below 37 Saybolt seconds. Plot specific gravity vs. temperature on its grid.

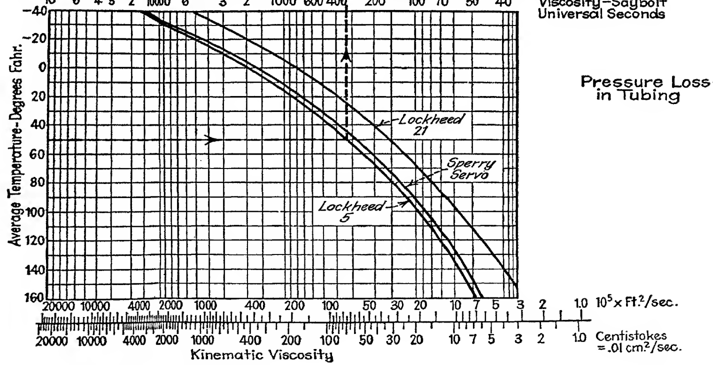


FIG. 16.—Pressure loss in tubing.

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First, it must be found whether these particular conditions produce laminar or turbulent flow. This can be done by calculating the Reynolds number, which will show whether the flow is laminar or turbulent. If R is above 2000, the flow is turbulent; if below, laminar.

$R = Vd\rho/\mu$. Since absolute viscosity μ is in poises, which is in c.g.s. units, and since in the determination of R the units must be consistent, conversion factors must be applied to convert V and d into c.g.s. units.

The calculation of the example now proceeds as follows:

$$V = \frac{4 \times 231 \text{ (cu.in./gal.)}}{1^2 \times 0.7854 \times 0.3937 \text{ (to convert to cm.)} \times 60 \text{ (to convert to seconds)}} = 49.8 \text{ cm./sec.} \; (= 1.63 \text{ ft./sec.})$$

$$d = 1 \times 2.54 \text{ (to convert to cm.)} = 2.54 \text{ cm.} \; (= 0.083 \text{ ft.})$$

$$\rho = 1 \text{ (water, gm./cu.cm.)} \times 0.87 \text{ specific gravity} = 0.87 \text{ gm./cu.cm.}$$

$$\mu = 0.296 \text{ poise}$$

$$R = \frac{Vd\rho}{\mu} = \frac{49.8 \times 2.54 \times 0.87}{0.296} = 372$$

Since R is under 2000, the flow must be laminar. In calculating the value of f, the formula for laminar flow is used.

$$f = \frac{64}{R} = \frac{64}{372} = 0.172$$

By using these values in the formula for head loss, there is obtained

$$h = f\frac{V^2L}{d2g} = \frac{0.172 \times 1.63^2 \times 15}{0.083 \quad 64.4} = 1.3 \text{ ft. head lost}$$

which agrees with the value obtained from the chart within the usual limits of accuracy of such charts.

Fluid resistance may be obtained from the chart in two forms, as head loss and as pressure loss. These terms have the same meaning, for pressure within a fluid column is developed by the weight of the fluid head above the point where the pressure is measured. Pressure is a more convenient measuring unit for aircraft hydraulic

work, however, for a pressure of 1 lb./sq. in. will always do the same amount of work, whereas the work that can be done by a head of 1 ft. depends on the weight of the fluid used. Because the pressure is equal to the weight of a column of fluid above the point where the pressure is measured, it can be seen that pressure $p = \rho h$ where all the units are in a consistent system, as p in pounds per square inch = weight in pounds per cubic inch × head in inches. For water, p in pounds per square inch = 0.0363 × head in inches, or 0.435 × head in feet. For any other fluid, $p = 0.435 \times$ head in feet $\times s$, where s is the specific gravity, which is the ratio of the weight of the fluid to the weight of an equal volume of water.

Of course, in an actual airplane system pressure is never produced by a head of fluid but rather is the result of mechanical force such as the pressure of a piston on a body of fluid in a pump. In this case, "head" means the height to which fluid would rise if a sight gauge were to be put on a pressure line, which is usually such a great height that the pressure could not possibly be measured in this manner. The only occasion for using head as such is in suction lines to pumps, where the height of the reservoir above the pump may produce an appreciable pressure at the pump inlet.

Because it is often used in hydraulic textbooks, it has been included here; but, in practical work, pressure drop in lines is read directly from the chart (Fig. 16).

To sum up briefly the subject of fluid resistance due to flow in pipes, it can be said that the resistance is due to the shearing between adjacent parts of the moving fluid, which may be in a laminar or turbulent state of flow; that the resistance depends on the viscosity of the fluid; and that this resistance can be calculated but can be much more easily determined from a chart constructed to cover the ranges of flow, pipe size, and viscosity most often encountered in airplane hydraulic systems.

FLOW THROUGH ORIFICES AND AT CHANGE OF SECTION IN PIPE

Resistance to fluid flow in hydraulic systems can be created in two ways, by resistance to flow in pipes caused by the shearing action of flow in a viscous fluid, and by turbulence, causing energy loss at changes of section in pipes or at orifices. To understand how resistance can be created, *i.e.*, how head can be lost, at a

change of section, it is necessary to consider what happens when the fluid flows past certain types of changes in section.

First, however, a few fundamental concepts must be clearly understood, as follows: (1) the meaning of energy, and the difference between potential and kinetic energy; (2) the fact that in ideal fluid flow, in which there are no losses, the sum of potential and kinetic energies, *i.e.*, the total energy, must be the same throughout the entire system. This latter principle is known as that of the *conservation of energy* and, in its application to hydraulics, as *Bernoulli's theorem,* after the Swiss who first applied this law to hydraulics.

Energy is the capacity for doing work and is measured in the same terms in which work is measured. If a force of 1 lb. moves through a distance of 1 ft., 1 ft.-lb. of work is done. From this it follows that work is equal to a force times the distance through which it moves and is expressed in force times distance units. If a 1-lb. weight is suspended 1 ft. above the floor, it is capable of doing 1 ft.-lb. of work before it comes to rest. Since this is capacity for doing work, it is called *energy;* and since it is at rest, it is called *potential energy*, to distinguish it from kinetic energy or energy of motion. Thus, the 1-lb. weight suspended 1 ft. above the floor is said to have a potential energy of 1 ft.-lb.

If the same 1-lb. weight is rolled along the floor at a certain speed, it will have no potential energy since it cannot fall any distance; but since it obviously has the capacity for doing work, it must contain another kind of energy. This is *kinetic energy*, or *energy of motion.* This energy can be transformed into work by directing the moving weight against some object requiring force to move. While the moving weight is decelerating, it will continue to give up energy and do work until it comes to a standstill, when it will have no remaining capacity for doing work, *i.e.*, no potential or kinetic energy.

The kinetic energy in the moving weight can be transformed into potential energy by guiding the moving weight up a track, which will cause it to slow down. When it has stopped, all its kinetic energy has been transformed into potential energy. In reversing the process, where a 1-lb. weight is released 1 ft. above the floor there are no appreciable losses involved, and all the potential energy will be converted into kinetic energy at the time the floor level is reached. From this the conclusion can be drawn that the kinetic

energy of a body moving at a certain velocity can be expressed as the weight of a body times the height from which it would have to fall to produce that velocity.

The applicable formula, which can be found in any physics textbook, is $h = V^2/2g$.

Upon putting in the weight factor, which normally is canceled out because in the case of fluid flow the flow is the same at any point and therefore W (in unit time) is the same weight at any point and may be canceled out, there is obtained Wh (= potential energy) = $WV^2/2g$ (= kinetic energy). It also appears that when there are no losses the sum of the potential and kinetic energies is always constant, which is the principle of conservation of energy.

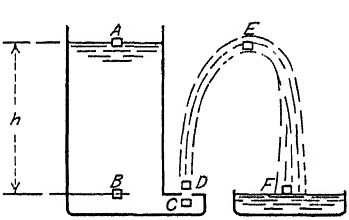


Fig. 17.—Flow through orifices.

With these fundamental principles in mind, the subject of flow through orifices and at local changes of section in pipes will now be discussed. The simplest case, that of flow from the orifice into the open air, is exemplified in Fig. 17. A particle of fluid at A has stored up in it potential energy equal to its weight times the distance it can fall, h. When the particle of fluid reaches B, it being assumed that the head of fluid above it is still equal to h, the potential energy due to its height has been transformed to a different form of potential energy, in this case pressure, due to the weight of fluid above it. A particle of fluid at C, just entering the orifice, also has a pressure potential energy in it equal to its weight times h.

However, as soon as it is outside the orifice, at D, it can have no pressure, since it has no restraining walls, but now is moving fast and has its energy in the form of kinetic energy. Its velocity at this point can be found by rearranging the formula $h = V^2/2g$ to get $V = \sqrt{2gh}$.

By the time the particle of water has reached E, all its kinetic energy has been converted into potential energy, which can be seen to be the same, on the assumption of no losses, of course, as that of a particle at A. In falling from E to F the potential energy is changed again to kinetic energy. But now a different factor comes into play; and by the time the particle, which had all its original energy remaining at F, has come to rest in the tank into

which the jet is falling, all its original energy has been dissipated, since it has no height or velocity. This energy is dissipated by turbulence causing local shearing forces to be set up between adjacent particles of fluid and thus causing fluid resistance with consequent loss of energy. This energy is not actually lost but is converted into heat and thus cannot be used to do useful work, in the sense in which the term "useful work" is used in hydraulics, *i.e.*, mechanical work.

At any point between those noted on Fig. 17 the energy of the particle of fluid is in part potential and in part kinetic.

It has been seen that the velocity of flow through an orifice $= \sqrt{2gh}$. The quantity of fluid that will flow out the orifice in a given time depends on the area of the stream and is obtained by multiplying the area of the stream by the velocity. When Q is the quantity of fluid flowing in a given time, the formula for discharge is $Q = A$ (stream) $\times V$. If all the terms in the equation are given in a consistent system of units, the discharge will be in those units; for instance, if foot-pound-second units are used, the result will be in cubic feet per second.

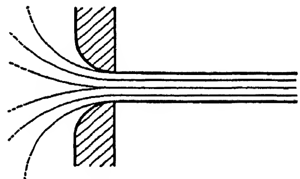


FIG. 18.—Rounded orifice.

But the area of the stream is not necessarily the same as the area of the orifice. If the orifice is smoothly rounded on the entering side so that the flow lines are straight after leaving the orifice as in Fig. 18, the stream is the same size as the orifice, and the formula previously given may be used, with A as the area of the orifice as well as the area of the stream, and with the theoretical $\sqrt{2gh}$ substituted for V, giving $A\sqrt{2gh}$. If the orifice is of any other shape, however, the area of the stream will be different from the area of the orifice and the velocity may not equal the theoretical because of losses.

The difference between the area of the orifice and the area of the stream is taken care of by introducing a *coefficient of contraction.* This coefficient is the ratio of the area of the stream to the area of the orifice. It is called the coefficient of contraction because of the contracted or reduced section of a stream issuing from a sharp-edged orifice, as in Fig. 19. This contraction is caused by the inflow velocity of the fluid approaching the orifice.

The difference between the velocity of the fluid leaving the orifice and the theoretical velocity is taken care of by introducing a *coefficient of velocity*. This coefficient is the ratio of the actual to the theoretical velocity. The reduction in velocity is caused by drag, or fluid resistance, in the orifice and is found whenever there is fluid resistance due to excessive wetted surface in the orifice or wherever there is turbulence within the orifice. A typical case of

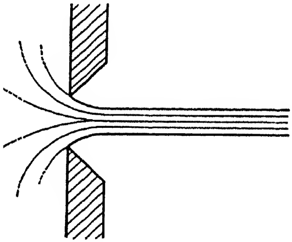


FIG. 19.—Sharp-edged orifice.

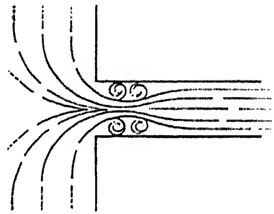


FIG. 20.—Short-tube orifice.

reduction in velocity is encountered in a short tube in which turbulence causes fluid resistance, which results in loss of energy with resulting reduction in velocity, as in Fig. 20.

Since the discharge through an orifice equals the actual area of the orifice multiplied by a coefficient of contraction, times the theoretical velocity multiplied by a coefficient of velocity, both of which coefficients depend on the physical characteristics of the orifice, it is evident that the formula can be simplified by multiplying the coefficients of contraction and velocity to get a coefficient of discharge for any orifice. When this is done, the commonly used formula for discharge through an orifice results.

$$Q = C_D \times A \text{ orifice} \times \sqrt{2gh}$$

This formula can be used with any consistent system of units.

In flow wholly within a pipe, the same principles apply as those which apply to a free jet, but they are perhaps more difficult to visualize. In considering a pipe having a reduced section with no abrupt changes in section so that no turbulence losses occur (the losses due to fluid resistance in the pipe can be calculated separately), it is seen from the principle of conservation of energy that since the velocity is increased at the reduced section, and therefore the kinetic energy is increased (since the quantity of

fluid flowing is the same), the pressure in the reduced section must be correspondingly decreased to reduce the potential energy and keep the total the same as the total energy in the pipe outside the reduced section (see Fig. 21). If, however, the expansion, instead of being accomplished gradually, without losses, is accomplished suddenly, turbulence and therefore fluid resistance occurs. The energy loss that occurs results from the slowing down of the

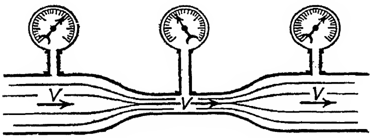


FIG. 21.—Flow through pipe at reduced section (gradual contraction and gradual expansion).

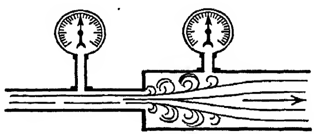


FIG. 22.—Flow at abrupt exit from reduced section (sudden expansion).

fluid stream without a compensating increase in pressure (see Fig. 22).

Since the kinetic energy in the stream and hence the energy loss is independent of viscosity, the fluid resistance due to flow past sudden change of section in pipes differs from fluid resistance due to flow in straight pipes in that viscosity does not enter into the calculation of the resistance and therefore the resistance measured as head loss is independent of temperature, although the pressure loss varies with the specific gravity, which varies slightly with temperature. Except in special cases, it is sufficiently accurate to take an average specific gravity and neglect the effect of change of specific gravity with temperature.

To return to the case of energy loss due to sudden expansion, the kinetic energy in the reduced section, in which the fluid is flowing at a velocity V_1, is $V_1^2/2g$ and the kinetic energy in the enlarged section after the sudden expansion has taken place is $V_2^2/2g$. The energy loss is the square of the difference in velocity, or

$$\frac{(V_1 - V_2)^2}{2g}$$

The effect of the energy loss on the pressure in the system is to reduce the pressure in the pipe after the sudden expansion below what it would be if there were no energy loss. It therefore represents a pressure drop that must be added to the pressure drop

resulting from flow resistance to get the total pressure drop in a line having changes in section.

The amount of this change in pressure can be calculated by first calculating the pressure in the pipe, on the assumption of no energy loss, and then subtracting the pressure lost due to turbulence. The pressure in the pipe after the expansion, on the assumption of no losses, can be calculated by remembering that the potential-energy pressure increase must just balance the kinetic energy decrease since there is the assumption of no losses. Thus

$$P_1 + \frac{V_1^2}{2g} \qquad P_2 + \frac{V_2^2}{2g} \quad \text{or} \quad P_2 = P_1 + \frac{V_1^2 - V_2^2}{2g}$$

Then the energy loss is calculated by the formula

$$P = \frac{(V_1 - V_2)^2}{2g}$$

with the resulting final pressure in the enlarged section after the sudden expansion equaling

$$P_1 + \frac{V_1^2 - V_2^2}{2g} \qquad \frac{(V_1 - V_2)^2}{2g}$$

If P_1 is omitted, the remainder of the equation is the change in pressure resulting from the flow from a small pipe past a sudden expansion into a larger pipe. In these equations, P is in head, and a conversion factor must be applied to get the result in pressure.

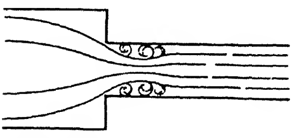


FIG. 23.—Flow at abrupt entrance to reduced section (sudden contraction).

Of the four possible cases of change in section, gradual expansion, gradual contraction, sudden expansion, and sudden contraction, the first three have been considered. For the case of sudden contraction, see Fig. 23.

There is no loss resulting directly from the contraction, since the flow lines are smooth and without turbulence; but there is a loss resulting from the sudden expansion following the contraction due to the inflow of the fluid approaching the orifice. This has been experimentally determined for various ratios of diameter of large to small pipe and will be plotted on the following charts. For entrance from a large vessel into a pipe the *energy loss* is half the

loss that would result if the flow were reversed at the same velocity of flow in the pipe; it is represented by the formula, for this specific case, loss $= 0.5(V_1 - V_0)^2/2g$. Since V_0, the velocity in the large vessel, $= 0$, the formula reduces to the form $0.5V^2/2g$.

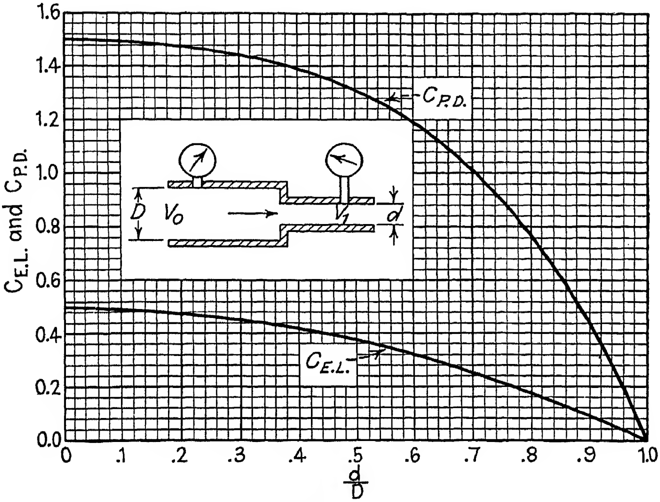


FIG. 24.—Pressure and energy changes, sudden contraction in a pipe.

$$\text{Energy loss} = C_{EL} \times \frac{V_1^2}{2g}$$

$$C_{EL} = 0.5 - 0.5\left(\frac{d}{D}\right)^2 \text{(empirical)}$$

$$\text{Pressure (head) decrease} = \frac{V_1^2 - V_0^2}{2g} + C_{EL}\frac{V_1^2}{2g} = C_{PD}\frac{V_1^2}{2g}$$

Any consistent system of units may be used.
When ft.-lb.-sec. units are used, result is in feet head of fluid.

To get the *pressure change* for the case of entrance to a pipe, the pressure drop due to the above energy loss must be added to the pressure drop (on the assumption of no losses) resulting from speeding up the flow. This is the same procedure as that followed in the case of sudden expansion except that, in this case, pressure reductions are added instead of a pressure reduction being subtracted from a pressure increase. The resulting formula is

$$P(\text{reduction}) = \frac{V_1^2 - V_0^2}{2g} + \frac{0.5(V_1 - V_0)^2}{2g}$$

If instead of a large vessel a larger pipe is used, for which the ratio of diameters has a finite value, the formula has the same form except that the coefficient 0.5 is different for every ratio of diameters, approaching 0 (*i.e.*, less turbulence) as the diameters approach one another.

In the calculation of the losses due to changes of section in a hydraulic system, either energy loss or pressure change can be

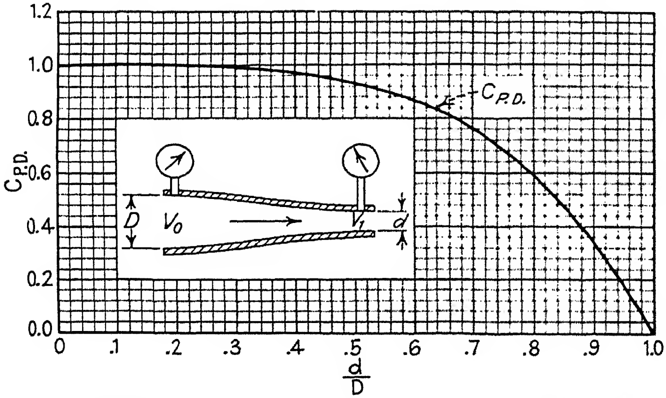


Fig. 25.—Pressure and energy changes, gradual contraction in a pipe.

Energy loss = 0

$$\text{Pressure (head) decrease} = \frac{V_1^2 - V_0^2}{2g} = C_{PD}\frac{V_1^2}{2g}$$

Any consistent system of units may be used.
When ft.-lb.-sec. units are used result is in feet head of fluid.

used, with the same final result provided that the same method is followed throughout the entire system.

To reduce the required calculation to a minimum the charts in Figs. 24 to 28 have been plotted. In constructing these charts all the necessary corrections have been applied to the coefficients so that to get the energy loss or pressure change at change of section it is necessary only to select the proper coefficient, from consideration of the type of change of section and the ratio of the diameters or areas, and multiply by $V_1^2/2g$ where V_1 is always the velocity at the small section.

The multiplying factor has been kept as $V^2/2g$ so that any consistent systems of units may be used. The result, of course, comes out as feet head, which must be converted to pressure. By using

Fig. 28 in combination with Figs. 24 to 27 the pressure drop can be found without calculation. Figure 28 is a chart for finding the

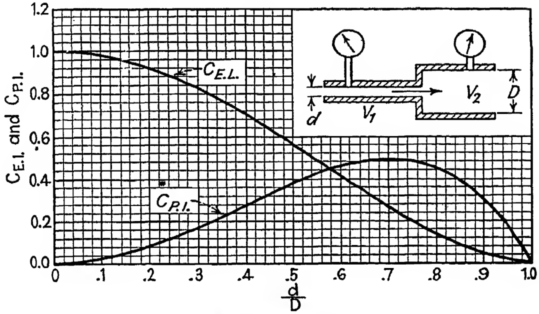


FIG. 26.—Pressure and energy changes, sudden expansion in a pipe.

$$\text{Energy loss} = \frac{(V_1 - V_2)^2}{2g} = C_{EL}\frac{V_1^2}{2g}$$

$$\text{Pressure (head) increase} = \frac{V_1^2 - V_2^2}{2g} - \frac{(V_1 - V_2)^2}{2g} = C_{PI}\frac{V_1^2}{2g}$$

Any consistent system of units may be used.
When ft.-lb.-sec. units are used result is in feet head of fluid.

pressure drop through an orifice for a coefficient of 1.00 when the flow and the orifice size are known; *i.e.*, it is a plot of $V^2/2g$ including the proper correction factors to get the result in pounds per square inch pressure drop. Since Figs. 24 to 27 give the coefficient

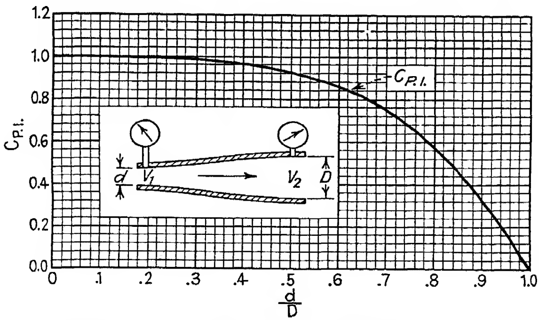


FIG. 27.—Pressure and energy change, gradual expansion in a pipe.

$$\text{Energy loss} = 0$$

$$\text{Pressure (head) increase} = \frac{V_1^2 - V_2^2}{2g} = C_{PI}\frac{V_1^2}{2g}$$

Any consistent system of units may be used.
When ft.-lb.-sec. units are used result is in feet head of fluid.

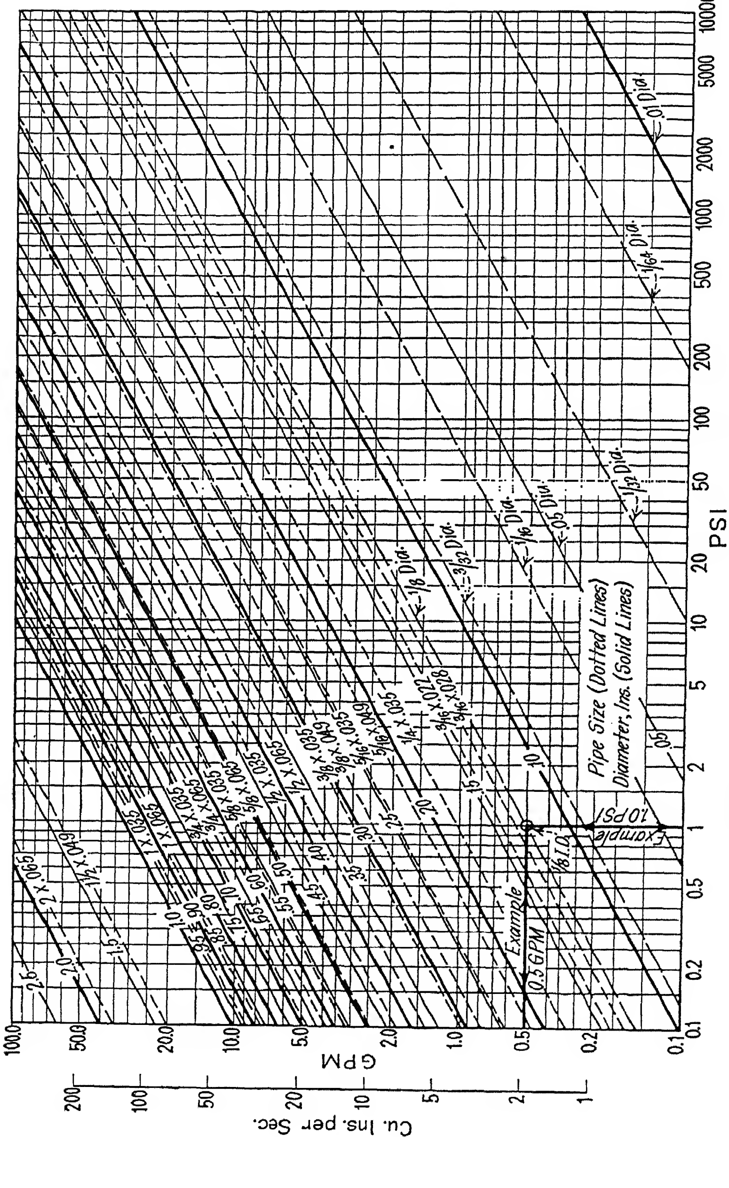


FIG. 28.—Discharge through an orifice for a coefficient of 1.00.

1. Discharge from an orifice.

Chart gives discharge from orifice with coefficient of discharge = 1.0. Correct for other coefficients as follows:

When solving for GPM:

Actual GPM = chart GPM × coefficient of discharge.

When solving for PSI:

$$\text{GPM to use on chart} = \frac{\text{actual GPM}}{\text{coefficient of discharge}}$$

2. Pressure change at change in pipe cross section.

Solving for PSI:

a. Given GPM and diameter of smaller pipe, find pressure change (see example).
b. Multiply this value of PSI by proper coefficient as determined from charts on preceding pages.

Solving for GPM:

a. Divide actual pressure change by proper coefficients as determined from charts on preceding pages.
b. Use this value and diameter of smaller pipe to find GPM (see example).

Solving for d = diameter of smaller pipe:

a. Assume d/D ratio and determine coefficient from charts on preceding pages.
b. Divide actual pressure change by this coefficient.
c. Use this PSI and actual GPM to find d.
d. Check d/D ratio and repeat process if necessary.

3. Specific gravity correction.

Chart is based on S.G. = 0.87 (Sperry oil at 70° F.)
Correct for other specific gravities as follows:

When solving for PSI:

$$\text{Actual PSI} = \text{PSI from chart} \times \frac{\text{actual S.G.}}{0.87}$$

When solving for GPM:

$$\text{PSI to use on chart} = \text{actual PSI} \times \frac{0.87}{\text{actual S.G.}}$$

for flow past various types of change of section, the actual pressure drop in pounds per square inch can be found by multiplying the pressure drop from Fig. 28 by the proper coefficient from Figs. 24 to 27. As an example, if it is desired to find the energy loss at the entrance to a pipe ½ in. outside diameter × 0.035 in. wall, with a flow of 12 g.p.m., it can be calculated from Fig. 24 (sudden contraction) that the energy loss will be

$$C_{EL}\,\frac{V_1^2}{2g} = 0.5\,\frac{(26.4)^2}{2 \times 32.2} = 5.4 \text{ ft. head}$$

and that the reduction in pressure will be

$$C_{PD}\,\frac{V_1^2}{2g} = 1.5\,\frac{(26.4)^2}{2 \times 32.2} = 16.2 \text{ ft. head}$$

When using Fig. 28, trace from 12 g.p.m. in the left-hand column over to ½ × 0.035 and down to a pressure drop ($c = 1$) of 4.08 lb./sq. in. Then the energy loss will be 0.5 × 4.08 = 2.04 lb./sq. in., which agrees with the calculated figure of 5.4 ft. head, and the pressure reduction will be 1.5 × 4.08 (from Fig. 28) or 6.12 lb./sq. in., which agrees with the calculated figure of 16.2 ft. head. It must be understood that to obtain the total energy in foot-pounds lost in a given space of time the energy loss expressed as head loss must be multiplied by the quantity of fluid flowing. In the case of a head loss of 2 ft. and a flow of 1 cu. ft. (62.5 lb.) of fluid, in this case, water, the energy loss is 2 × 62.5 = 125 ft.-lb. If the loss is expressed as pressure in pounds per square inch, which in this case would be 0.87 pounds per square inch, then the result is 0.87 lb./sq. in. × 144 (to get pounds per square foot) × 1 cu. ft. = 125 ft.-lb. energy loss.

However, it is more convenient to work with energy loss and pressure reduction expressed as feet-head or pounds per square inch rather than foot pounds since these former values are independent of flow.

It has already been shown by an example how the line-loss chart is used in determining the pressure drop in a line in an airplane hydraulic system. The use of the orifice charts in determining the pressure drop in fittings is somewhat more complicated. First a sketch of the fitting is drawn, as, for example, the check valve of Fig. 29. In this check valve the flow consists of a sudden expan-

sion at *a*, gradual contraction at *b*, sudden expansion at *c*, gradual contraction at *d* (no turbulence and, therefore, no loss occur at entrance since the plate is so thin; all loss is taken care of in condition *e*), and sudden expansion at *e*.

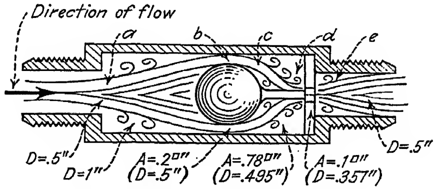


FIG. 29.—Flow in check valve.

The calculation would be as follows: Flow is assumed 12 g.p.m.; and diameters are as noted in Fig. 29. (Note that areas have been converted to diameters of the same area to facilitate use of the charts, which are plotted as diameters.)

Point	Condition	d/D	C_P, Figs. 24–27, + increase − decrease	Pressure $C = 1$, lb./sq. in., Fig. 28	Pressure change, lb./sq. in., + increase − decrease
a.	Sudden expansion.	0.5	+0.38	2.25	+0.85
b.	Gradual contraction. . . .	0.5	−0.93	2.25	−2.09
c.	Sudden expansion.	0.503	+0.38	2.25	+0.85
d.	Gradual contraction. . .	0.36	−0.98	9.30	−9.13
e.	Sudden expansion.	0.73	+0.50	9.30	+4.65
	Total pressure drop, lb./sq. in.	. . . .	. . . .	. . . .	4.87

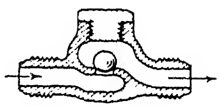
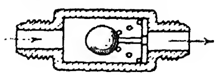
The viscous resistance has been neglected since the pipe length used in calculating the viscous resistance includes the fittings. Viscous resistance cannot be neglected if a fitting having a reduced section of appreciable length is used, however, as the resistance of the high-velocity flow will be greater than the resistance of an equal length of the basic pipe size.

Thus, the pressure drop through this fitting is 4.87 lb./sq. in. at 12 g.p.m.; and since V is directly proportional to flow and pressure drop varies with V^2, this pressure drop will vary with g.p.m.^{2} just as in an orifice.

For handy calculation, this fitting can be replaced with an orifice having the same pressure drop and flow characteristics. From Fig. 28 the orifice having a pressure drop of 4.87 lb./sq. in. at 12 g.p.m. is 0.42 in. in diameter. Therefore, in all future designs in which this check valve is used, it is only necessary to add the loss from Fig. 28 for a 0.42 in. orifice to the other pressure losses to account for the loss through the check valve. After a little experience a table of "equivalent orifices" will have been worked up by calculation or test for the most commonly used check valves, elbows, and similar fittings, from which values can be taken for calculations for any new system. From inspection of the table, the equivalent orifice for new fittings or fittings of unknown internal dimensions can be estimated with fair accuracy. The accompanying Table I is such a table.

TABLE I.—EQUIVALENT ORIFICES FOR HYDRAULIC UNITS

(To be used with Fig. 28 in determining pressure drop through units)

Unit	Size, in. (O.D. of connecting tubing)	Equivalent orifice, in. (diameter)	Remarks
Ball check (globe type)..	1/4	0.090	
	3/8	0.135	
		0.180	
Ball check (straight flow)	1/4	0.175	
	3/8	0.260	
		0.350	
Poppet 4-way valves....	1/4	0.077	For flow from "Pressure" to "Cylinder" or "Cylinder" to "Return"
	3/8	0.120	
	1/2	0.154	
Lapped slide valves....		0.125	For flow from "Pressure" to "Cylinder" or "Cylinder" to "Return"
	3/8	0.187	
	1/2	0.250	

To sum up briefly the subject of fluid resistance due to flow through orifices and changes of section, it can be said that the resistance is due to the turbulence resulting when the flowing fluid

is slowed down suddenly, that it is independent of temperature, and that it can be calculated but can be much more easily determined from a table of equivalent orifices constructed to cover the types of fittings most often encountered in airplane hydraulic systems, together with a chart for determining the pressure drop through these equivalent orifices at various flows.

CHAPTER III

HYDRAULIC SYSTEMS

DEVELOPMENT OF THE MODERN HYDRAULIC SYSTEM

In Chap. I, a hydraulic system was defined and a basic system was described, consisting of a reservoir, pump, valves, cylinder, and lines connected as shown in Fig. 30.

Hydraulic systems in modern airplanes are far more complex than this basic system, but their development has proceeded by logical and necessary steps from the basic system. Probably the best way to understand the modern airplane hydraulic system is to follow its development from the basic system.

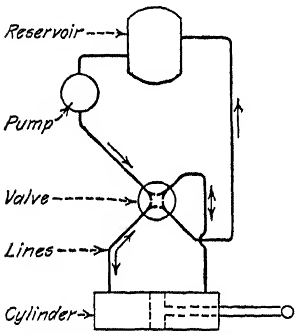


FIG. 30.—Schematic diagram of basic airplane hydraulic system.

Airplanes of a few years ago, when they had a hydraulic system at all, used a system almost identical with the system shown in Fig. 30. That is, they had a fluid-supply reservoir, a pump, usually manually operated, a valve, and one or, if a landing gear was to be retracted, usually two operating cylinders. Thus the first addition to the basic system that was found necessary was the addition of another cylinder, so that retractable landing gears could be conveniently operated.

Such a system is shown in Fig. 31. As shown by the dotted lines and cylinders, any number of cylinders can be added on one valve circuit as long as they are added in parallel rather than in series. Since the pressure is equal throughout a hydraulic system, except for line loss, which is small compared with the pressure available, the unit that requires the lowest pressure to move will operate first; in fact, if it continues to have the lightest load, it will continue to move until it reaches the end of its travel, when the pressure will build up until it reaches a value sufficient to move the

next most lightly loaded cylinder, when this cylinder will start to move. This process of increasing pressure continues until the cylinder that requires the most pressure to make it operate has reached the end of its travel.

Thus it can be seen that a system of cylinders in parallel is not synchronized and the sequence of operations is determined by the

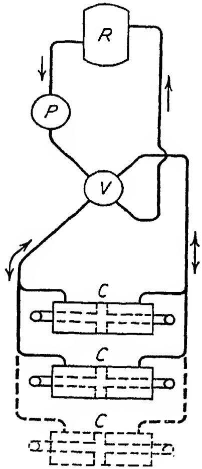


FIG. 31.—Schematic diagram showing addition of cylinders in parallel to basic airplane hydraulic system.

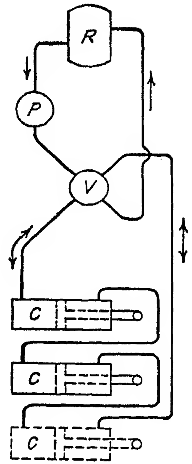


FIG. 32.—Schematic diagram showing addition of cylinders in series to basic airplane hydraulic system.

loads on the cylinders. This explains why, in landing gears retracted hydraulically, one wheel often retracts completely before the other starts to retract. The wheel that retracted first had less air load, less bearing friction, or a slightly lower weight or for some reason required a lower pressure to operate.

If cylinders are hooked up in series, as in Fig. 32, rather than in parallel, an entirely different condition results. Fluid is now trapped between the cylinders with no possible way to return to the reservoir. When the valve is operated, this trapped oil will be forced from the first cylinder into the second and from the second into the third, this process ensuring that all three cylinders, if they are of the same area, move at the same rate. Unfortunately the trapping of fluid between one cylinder and the next introduces

serious complications into the design of the system. A leakage, however slight, of the fluid trapped between the cylinders will result in the cylinders getting out of synchronism, for when the first cylinder has reached the end of its travel the second cylinder has not quite reached the end of its travel owing to the smaller quantity of trapped fluid remaining after leakage. A similar problem, but in the reverse order, occurs when the trapped fluid expands as a result of an increase in temperature. These problems can be overcome by the use of synchronizing valves arranged to

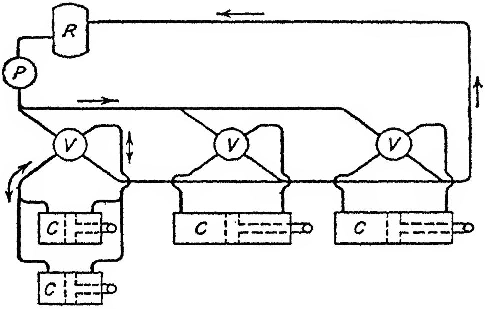


FIG. 33.—Schematic diagram showing addition of valves to basic airplane hydraulic system.

open automatically when the cylinders reach the ends of their travel, thus permitting a flow of fluid to replace leakage.

A further disadvantage of series cylinder arrangements is excessive weight, for each cylinder must be designed for the full displacement rather than the displacement being divided between cylinders as is the case in parallel arrangements. This results from the fact that in series arrangements the pressure, rather than the volume, is divided between the units in the system.

Because of these disadvantages, the series hookup is used only where absolutely necessary. The details of the designing of series systems are covered in a later chapter (Chap. VIII) on Design of Special-purpose Units.

The next addition to the hydraulic system after the use of more than one cylinder on one valve circuit came when it was found desirable to use the same hydraulic system to operate more than one service, as, for example, landing gear and wing flap. This requires more than one valve on the same power circuit, *i.e.*, on

the same pump or pumps. A system using more than one valve is shown in Fig. 33.

Obviously, if all valves except one are kept shut off, the addition of the valves does not change the operation of the system in any way over the hydraulic systems previously described. If, however, more than one valve is operated, then the system requiring the lowest pressure to operate will operate first and will continue to operate until it reaches the end of its travel or until the load on it increases above the load on some other cylinder. Thus the cylinders in a multivalve system, with more than one valve operated, act just as they would if they were all hooked up in parallel on one valve, as in Fig. 31.

Cases of hookup or operation other than those previously covered can all be solved by applying the following two general principles:

1. Flow in a hydraulic system always tends to equalize pressures, *i.e.*, to build up low pressures and lower high pressures.
2. Fluid is incompressible, and so the volume in a subsystem remains the same (plus pump input minus flow to reservoir).

Upon applying these to the case in which, after a highly loaded system has reached the end of its travel with its valve still open, the valve leading to a lightly loaded system is operated, it is seen

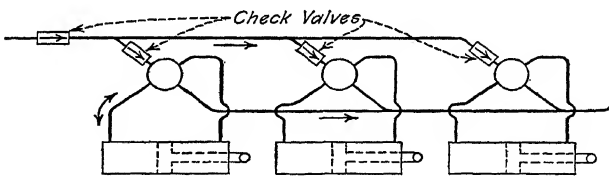


FIG. 34.—Addition of check valves to prevent interflow.

that the fluid will flow back out of the highly loaded system into the lightly loaded system until pressures balance or the end of the travel is reached. Often, check valves or similar means are incorporated to prevent interflow between valves, as in Fig. 34.

Whenever a hydraulic system is closed off for a sufficiently long time to allow the temperature of the fluid in the system to increase appreciably with increase in outside-air temperature, it becomes necessary to add some means of temperature compensation, such as spring-loaded expansion chambers or relief valves. Relief

valves are by far the most common means of temperature compensation and are used wherever the low pressure resulting from a temperature decrease, following an increase that has caused fluid to be relieved from the system, is not detrimental. Where the pressure must be kept up, as in parking airplanes equipped with hydraulic brakes, a spring-loaded expansion chamber is required.

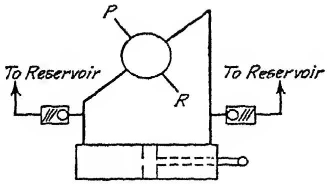


FIG. 35.—Use of relief valves in cylinder lines.

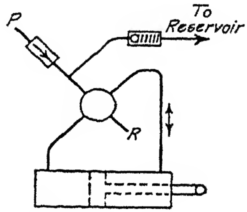


FIG. 36.—Use of relief valve in valve pressure line.

If the valve is kept in the off position, it is necessary to use a relief valve in each cylinder line, as in Fig. 35. If the valve is kept in its operating positions rather than in the off position, it is necessary only to provide a relief valve in the pressure line leading to the valve. This relief valve must be between the valve and any check valve that would prevent back flow out of the pressure line. If individual check valves for prevention of interflow between valves are used in front of each valve, then a separate relief valve is required for each directional control valve, as in Fig. 36. If only one check valve is provided into the common

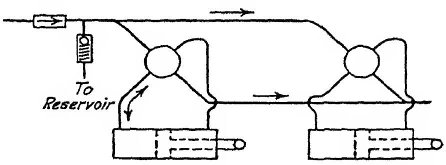


FIG. 37.—Use of relief valve in common pressure line.

pressure line, only one relief valve is required, relieving the common pressure line as in Fig. 37.

A fully developed airplane hydraulic system, from the common pressure line, or pressure manifold, onward, has now been described. Special-purpose units are often added, such as hydraulic landing-gear locks, special flap relief valves, etc., to perform special

functions. These units and their functions will be covered in the chapters dealing with the particular subsystem in which they are used. The system that has just been built up from the basic system contains only those units common to all subsystems.

DEVELOPMENT OF THE POWER SYSTEM

Next will be considered the part of the system that supplies pressure to the pressure manifold—the power system. On the early airplanes using hydraulic systems, the power system looked just as it does in the basic hydraulic system; *i.e.*, it consisted of a reservoir and a hand pump. As airplanes increased in size and speed, it soon became obvious that the $\frac{1}{20}$ to $\frac{1}{10}$ hp. which it was possible for the pilot to develop manually was not going to be sufficient to perform the necessary work, such as retracting the landing gear, operating the wing flaps, etc., in a reasonable time. The necessity for retracting the landing gear in a very short time immediately after take-off to reduce aerodynamic drag, in order to allow single-engine flight after failure of one engine on a bimotored airplane, also increased the horsepower required by the retractable landing gear on such airplanes.

The first step toward providing increased power was the installation of electric-motor-driven hydraulic pumps, usually of the gear type, in parallel with the hand pump, which was retained for emergency use. Various forms of control were used for the electric motors. Probably the simplest control was a push button, which had to be held on while the pump was running. In order to simplify the work of the pilot, most electric-motor-pump installations, however, used a pressure-actuated switch to cut off the motor when the cylinder reached the end of its travel and the pressure built up. Such a system is shown in Fig. 38.

In this system the pilot closed the switch to start the operation, and the pressure build-up at the end of the operation opened the switch. To prevent excessive pressure build-up after the switch opened, due to the inertia of the electric motor, which was sufficient to drive the pump several revolutions after the switch was opened, it was necessary to use a pressure relief valve in the pressure line near the pump outlet. This system proved in practice to have one serious disadvantage: it sometimes opened the switch before the operation was completed. This came about because

the resistance to flow was high when the oil was cold, and in cold weather the resistance to flow in the lines, plus the load from the unit being operated, built up sufficient pressure to open the switch before the operation was completed. Also, if an acceleration or "bump" was encountered while the gear was being retracted, the high pressure resulting from the high loads would operate the pressure switch.

There were two cures for this trouble, the use of lower viscosity fluid in winter to reduce the fluid resistance, and raising the kick-out pressure setting of the pressure switch. The use of thinner

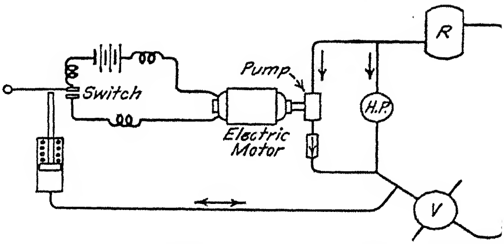


FIG. 38.—Use of electric-motor-driven hydraulic pump and pressure switch in power system.

fluid in turn had the disadvantage that it increased wear in pumps, etc., because of the poorer lubrication it provided. Increasing the pressure setting had many disadvantages. It shortened pump life, increased weight because of the stronger lines, cylinders, etc., required, and increased the maintenance cost of the system.

A fully satisfactory solution to the problem of control of electric-motor-driven hydraulic pumps was never found because with increasing weight and performance of airplanes the power requirements of the hydraulic system soon exceeded the electric power available, and to provide more electric power involved great weight increases.

The next type of system to come into use was the engine pump system, which is the system in use at the present time.

ENGINE PUMP SYSTEMS

There are three principal methods of pump control in use in present aircraft hydraulic systems, constant pressure, manual pump by-pass, and automatic pump by-pass.

In the constant-pressure system, as the name implies, the pump delivers fluid under full pressure at all times, excess fluid not required for operation of units being returned to the reservoir through a relief valve as in Fig. 39.

This type of pump control is suitable only for low horsepowers, for all the excess horsepower is transformed into heat in the relief valve, which, if it exceeds the radiating capacity of the system, results in excessive fluid temperature. The commonest example of such a system is the Sperry automatic pilot, in which the hydraulic horsepower dissipated is on the order of 1/5 hp.

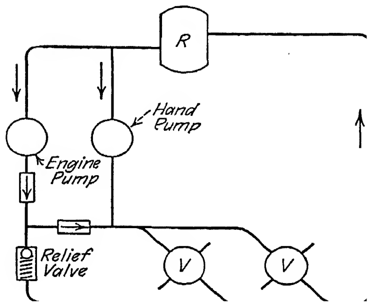


FIG. 39.—Schematic diagram of system using constant pressure method of pump control.

In all high-power hydraulic systems (with one exception) it is necessary to provide some sort of by-pass to allow the pump to circulate the fluid freely under no pressure except when a unit is actually being operated. This reduces the heating and pump wear to a minimum.

This by-passing can be done manually by several methods. The commonest of these involve the use of a pressure-operated by-pass valve, which is set manually and kicked out by the build-up of pressure at the end of the operation, or of a series of by-passes operated by the directional control valves so that when all the directional control valves are in neutral the by-pass is open, but the operating of any one of the directional control valves will close the by-pass.

The first of these methods, the manually closed by-pass valve, is shown in Fig. 40. The operation of a system using this method of pump control is as follows: When the pilot wants to operate a unit such as the retractable landing gear, he first moves the directional control valve and then closes the by-pass valve, thus forcing the entire pump output through the directional control valve and into the operating cylinders. When the operating cylinders reach the end of their travel, the pressure builds up and kicks the valve out. The directional control valve may be left in position or

returned to neutral when the operation is completed. This type of control has the same disadvantage as the pressure-operated switch control for electric-motor pumps previously described: *i.e.*, it can be kicked out before the operation is completed by high fluid resistance from cold oil or by high acceleration loads on the unit being operated. It does not suffer from cold-oil trouble to the same degree as the electric-pump system, however; for the constant circulation keeps the fluid warm in the circulating part of the system, including pump pressure and suction lines and the reservoir. This system has the disadvantages, when compared with the automatic-by-pass system, that it requires two motions on the part of the pilot to perform any operation, that it is liable to malfunction in cold weather or when accelerations are encountered, and that it is heavier because of the high ratio between normal working and maximum pressures. For these reasons, this type of pump by-pass is rarely used.

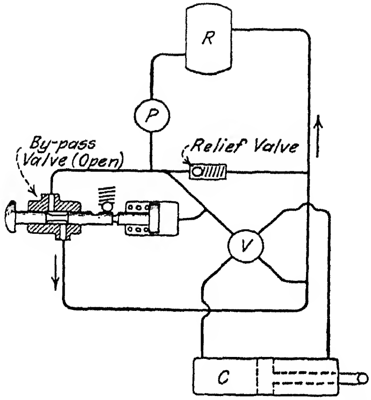


FIG. 40.—Schematic drawing of system using manually operated by pass valve.

The use of a series of by-passes operated by the directional control valves results in the system shown in Fig. 41. The operation of a system using this method of pump control is as follows: When the pilot wants to operate a unit, for instance, if he wants to retract the landing gear, he simply moves the directional control valve to the proper position, in this case the "Retract" position, waits until

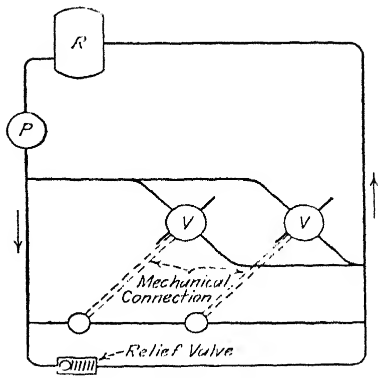


FIG. 41.—Schematic diagram of system using a series of by-pass valves operated by directional control valves.

the gear is retracted, and then returns the valve to neutral. When the valve is moved to "Retract," it not only directs the fluid to the proper cylinders, but also closes the associated by-pass valve. When the by-pass valve is closed, the fluid is no longer permitted to return freely to the reservoir but instead is forced to go into the operating cylinders and, after the pressure has built up at the end of the travel of the operating cylinders, out the relief valve. When the directional control valve is returned to neutral after the operation is completed, the by-pass is opened and the fluid again circulates freely.

This system is free from the disadvantages of the previous systems, but it has one disadvantage of its own. If the pilot fails to return the valve to neutral after the operation is completed, the fluid cannot by-pass freely but must go through the relief valve. This means that the entire pump output horsepower is converted into heat, which in the case of the modern airplane system is sufficient to heat the fluid to the point that the system becomes inoperative, either as a result of pump seizure or through rapid vaporization of the fluid. Because pilots have many duties besides operation of the hydraulic system, this mischance has proved to be of frequent occurrence, usually with resulting damage to the hydraulic system.

Several cures have been proposed, including time delay devices for returning the directional control valves to neutral after a reasonable time lag, pressure rise plus time delay devices, mechanical or electrical means for returning the valve to neutral by the last part of the travel of the unit being operated, and various other methods. These systems are at best added complications in an already complicated system and, when compared with a system using an automatic pump by-pass, present more disadvantages than advantages and therefore are now rarely used.

When the pilot wants to operate a unit in an airplane equipped with the automatic type of pump by-pass, such as the system diagramed in Fig. 42, he merely puts the valve in the proper position. Opening the valve drops the pressure in the pressure manifold, causing the by-pass to close and forcing the pump to pump fluid under pressure. When the pressure builds up at the end of the travel, the by-pass automatically opens, unloading the pump. If the fluid resistance is high, as in cold weather, so that when the full pump output is forced through the cylinder lines the pressure

exceeds the high limit, the by-pass opens and closes at short intervals, thus reducing the effective flow from the pump. In the case of a sudden accelerated load causing a temporary high pressure in the system the by-pass opens until the pressure drops to below the low limit and then closes again.

An accumulator is generally used with automatic pump-by-pass systems, for several reasons. One of these reasons is that it provides sufficient fluid to allow some leakage without causing

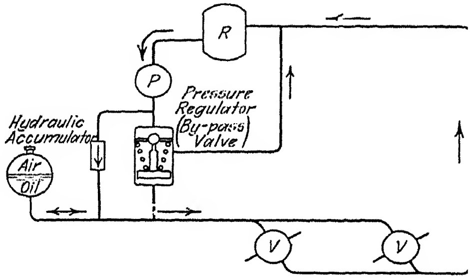


Fig. 42.—Schematic diagram of system using an automatic by-pass valve or pressure regulator.

extremely frequent operation of the automatic by-pass. Another is that it prevents shock pressures when the by-pass is opening, for without it pressures would rise very rapidly.

The detail hookup of the power system, including an automatic by-pass, will be discussed in the next chapter. Before leaving the subject of pump control, one system should be mentioned that, although it is not in wide use at the present time, may in the future prove very satisfactory. This is the use of automatic variable-displacement pumps. There are two forms of such control possible, variation in pump speed and variation in pump displacement. Of these, the pump in which the displacement, or quantity of fluid pumped per revolution, varies seems to be the most practicable and offers the most advantages. There are several such pumps on the market. The usual control arrangement is one that keeps the pressure in the pressure manifold constant within certain limits. When the pressure is at the low limit or below, the pump operates at full displacement. As the pressure approaches the high limit, the displacement is automatically reduced and reaches zero at the high limit. Some means for cooling the pump at zero

displacement is usually provided; for there is then no fluid flowing through the pump, which in constant-displacement pumps serves to carry off the heat generated by friction and internal leakage within the pump. This type of pump control has the advantage over the automatic by-pass that its operation is smooth and continuous rather than intermittent. This offers the advantages of noise reduction and elimination of shock pressures, both of which are important in very high horsepower systems.

The chief disadvantage at present is a lack of service experience with the pumps now on the market, and their relatively high cost, which is fundamental because of their large number of parts. Other disadvantages may be revealed as more such systems are put into operation.

Variation in pump speed has not yet been accomplished except by the use of a pressure-controlled clutch. This is not a desirable system, for it gives intermittent flow and subjects both pump and engine drive to severe shock loads.

PART II

DESIGN

CHAPTER IV

POWER SYSTEM

DESIGN REQUIREMENTS

In the previous chapter the steps leading to the use of automatic pump by-pass control have been described. Now there will be traced the development of the present power system using this type of pump control from the basic system of Fig. 30 to the systems in use at the present time.

Starting with a basic system, the first addition to the power system was that necessary to provide for increased power. This consisted of the addition of one or more power pumps, the automatic pump by-pass valve, and the necessary check valves to connect it into the system. In addition, a pressure-relief valve was usually added to take care of a possible failure to open of the by-pass valve and in general to act as a safety valve. In order to avoid getting air into lines when the engine section, including the pump, is removed from the airplane, it has become customary to add a valve at the fire wall, which automatically shuts off the pump line on both sides of the point where it is disconnected, in both the pressure and the suction lines. These valves are commonly known as "disconnect" valves.

When an accumulator is used, it is added on the pump line in such a way that it is kept charged with oil under pressure by the power pumps and discharges through a check valve into the pressure manifold.

Because the primary purpose of the hand pump in a power-pump system is to take care of emergencies, including failure of the power pump's automatic by-pass valve, the accumulator, or any other part of the power system, the hand-pump lines are usually arranged to discharge directly into the pressure manifold so that the fluid being pumped by the hand pump cannot be lost through broken lines in the power system. However, because it is desirable to be able to use the hand pump to charge the accumulator for operating parking brakes, etc., an accumulator charging valve is

usually incorporated in the system; this can be operated to allow the accumulator to be charged by the hand pump but is normally kept closed in flight. A diagram of the system incorporating the above units is shown in Fig. 43. A pressure gauge is also shown.

In Fig. 43 the fluid going to the engine pump comes out of the reservoir through a different outlet, at a higher level, than the

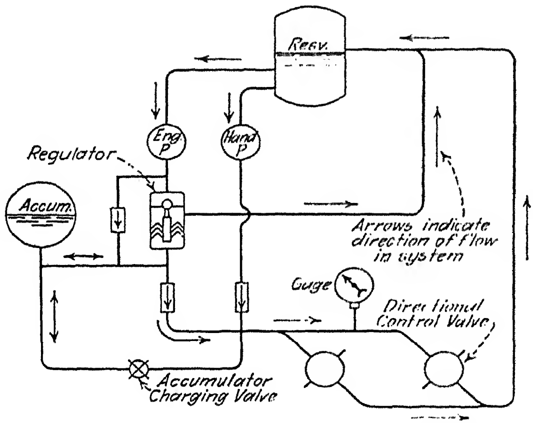


FIG. 43.—Schematic diagram of system incorporating accumulator charging valve.

fluid going to the hand pump. This is a valuable safety feature in that it permits use of the hand pump after a failure in the power-system lines.

Because airplane operators insist on longer and longer service from hydraulic units, it has become necessary to put filters into hydraulic systems to remove dirt particles, which might wear out pumps and valves. These filters may be located in the individual systems but are more commonly part of the power system. They may be of several types and may be in several locations, as shown in Fig. 44.

In Fig. 44, which is a diagram of a basic power system showing various filter locations, filter *A* is a filter in the suction line to the engine pump. This filter location has the advantage that it filters all the oil leaving the reservoir and provides the best possible protection for the power pumps. It has the disadvantage that it increases the pressure loss in the suction line, and to keep within

reasonable limits of suction at the pumps requires both very large filters and very large suction lines.

Filter *B*, in Fig. 44, is a filter in the return line of the circulating fluid. Considerably more pressure drop is available in this position, and a filter here does not detract from the performance of any subsystem; for there is no oil flowing in the circulating system, and therefore in the filter, at the time that the subsystems are being operated. A filter in this location filters the oil indirectly, rather than directly, and probably provides slightly less protection for the units in the system, particularly the engine pumps.

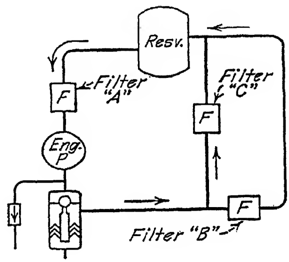


FIG. 44.—Schematic diagram of power system showing possible filter locations in the circulating system.

Filter *C* is a by-pass filter which should take off from some line having a reasonably high back pressure but should not, of course, take off from a pressure line, for that would detract from the performance of the hydraulically operated units.

If extremely fine filtration is desired and the system is one in which relatively large quantities of fluid circulate, the by-pass filter may be the only solution without resorting to excessively large filters in the other locations. The by-pass filter provides the poorest protection, for it filters only a small percentage of the total flow; however, it should eventually clean up the entire system. By-pass filters have proved satisfactory in protecting automobile engines.

Filters may be of various types. Reservoir filters are usually just screens, which may vary from 30 to 170 mesh. Occasionally, filters of the cleanable-blade type, such as Cuno filters, or filters of the wire-wound type, such as Purolater, are used in reservoirs. Circulating filters may be either the screen type of filters or may incorporate some nonmetallic filtering element, such as felt or cloth. By-pass filters are ordinarily used when extremely fine filtration is desired and therefore cannot be of the screen type. They usually consist of perforated cans packed with cloth fibers or layers of felt. The better types of such filters are usually built up in stages, a coarse stage removing the larger particles and one or more finer stages the finer particles.

When a hydraulic automatic pilot, such as the Sperry gyro pilot, is used, it may take its power from a separate pump and reservoir or it may take off from the airplane's hydraulic system. There are two common methods of taking the automatic-pilot fluid from the hydraulic system. The first of these is by means of a selector valve by which one of the power pumps of the airplane's hydraulic system is connected to the automatic pilot, the remaining pump or pumps being connected to the hydraulic system. The selector valve is incorporated in the system so that in the event of a failure of any one pump both the automatic pilot and the hydraulic system may be used on the remaining pump, although not, of course, at the same time. A power system incorporating this method of automatic-pilot power selection is shown in Fig. 45.

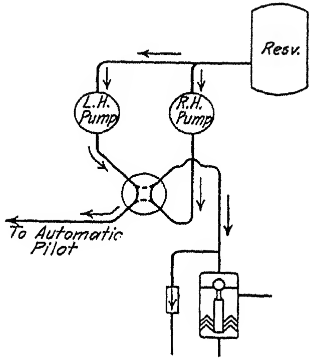


FIG. 45.—Schematic diagram of power system incorporating automatic pilot selector valve.

Fluid under pressure for operation of the automatic pilot may also be taken off the hydraulic system by means of a reducing valve, which reduces the fluid from the high pressure required to operate the airplane's hydraulic system down to the pressure required to operate the automatic-pilot system. The design of such units will be discussed in later chapters. This system is uneconomical if the hydraulic-system pressure is much higher than the automatic-pilot-system pressure, for considerable power loss and heating will result. This arrangement is shown in Fig. 46.

A variation of this arrangement, which is permissible in systems having relatively low flow, say up to 5 or 6 g.p.m. at cruising speed, is the use of a relief valve in the by-pass line from the engine pump or in the return line so that even when no unit is being operated there will be sufficient pressure for the automatic pilot, which requires about 100 to 150 lb./sq. in., backed up in the by-pass or return line. With this system the auto pilot will cease to operate whenever the accumulator is charging, for there will then be no fluid flowing in the by-pass or return line. To prevent this, a reducing valve hooked into the hydraulic pressure system

may be added. Such hookups are complicated and are not often used.

In the system as now designed, the unit requiring the least pressure will operate first. This sometimes makes for improper operation on the airplane; for example, if wing flaps requiring a large

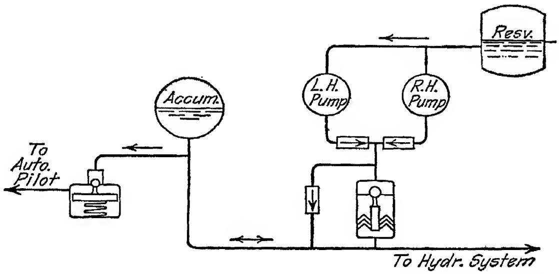


FIG. 46.—Schematic diagram of system incorporating reducing valve for autopilot operation.

quantity of fluid are to be raised immediately after landing, when the engine and therefore the hydraulic pump are operating at a low r.p.m., the brakes might become inoperative. It is possible, by adding relief valves to the system, to force the fluid into one system first until it builds up to a certain minimum pressure, after which it goes to other parts of the system. Thus it is possible to add a valve to cause the brake system always to have available at least 500 lb./sq. in., even, for example, when the flaps are being operated and are requiring only 50 lb./sq. in. This valve should not be the conventional relief-valve type, because if it were the pressure required to operate the wing flaps would have to be added to the pressure caused by the addition of this valve and this added pressure might overload the pumps; instead, it should be of the balanced relief-valve type shown in Fig. 47. With this valve the minimum pressure on the inlet side is always equal to the valve setting. As the pressure in the outlet side rises, it tends to relieve the spring load until, when the pressure in the outlet side is equal

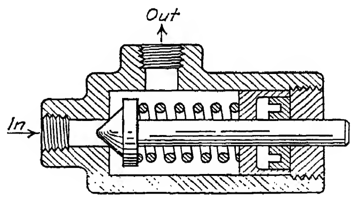


FIG. 47.—Schematic drawing of balanced relief valve.

to the pressure on the inlet side, the spring is completely backed off and there is free flow through the valve, causing no hindrance to operation. A system using a selective relief valve to ensure a supply of fluid under pressure to the brakes is shown in Fig. 48.

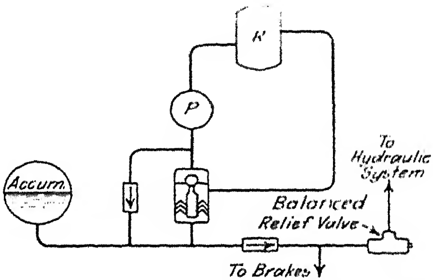


FIG. 48.—Power system incorporating balanced (selective) relief valve to insure pressure in brake system.

On airplanes intended to be operated in cold weather, it is sometimes desirable to keep the circulating fluid warmer than normal circulation will keep it; or, in order to effect rapid retraction immediately after take-off, it may be desirable to warm the fluid up rapidly. In these cases, it is possible to utilize the horsepower of the hydraulic system to warm up the fluid by forcing it to go through a relief valve or a restriction until it has attained the desired temperature. A diagram of a hydraulic system incorporating this means of temperature control is shown in Fig. 49.

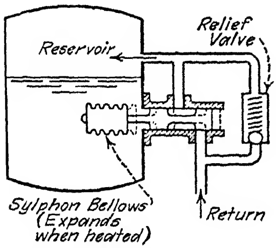


FIG. 49.—Schematic drawing of temperature-controlled system.

As airplanes increase in size and the horsepower of the hydraulic systems increases, the size and weight of the pump suction line increase disproportionately, for the pressure available to force fluid to the pump is only the atmospheric pressure available at the maximum altitude at which the airplane can fly. In modern airplanes this may mean an allowable pressure drop in the suction line of only 4 or 5 lb. sq. in.

To avoid excessively large suction lines, supercharged reservoirs are coming into use. These may be of the following two types:

either the completely closed type, in which a diaphragm or piston separates the fluid from a chamber filled with air under pressure; or the open type, in which air lost through the vent line when units are operated is replaced by some sort of air pump. A completely closed reservoir, such as the one shown in Fig. 50, has been used in airplanes requiring relatively low flows; however, there is some doubt as to its suitability for systems having high flows because of possible foaming of the fluid. It is also somewhat difficult to service a system equipped with such a reservoir.

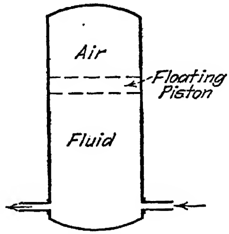


FIG. 50. — Schematic drawing of closed type of supercharged reservoir.

The other type of supercharged reservoir involves the use of a relief valve in the vent line, set to relieve at a pressure a few pounds above atmospheric, and a pump that will supply air under pressure, up to a pressure a little below the relief-valve setting. This system has also been used to date only on low-flow systems and may cause trouble on high-flow systems because of the tendency for air to become dissolved in the fluid. Experimental work is going on at the present time on both these systems, and probably one or the other will become the accepted standard for reservoir design in large high-altitude airplanes in the not too far distant future. Figure 51 shows the air-pump system of supercharging a reservoir.

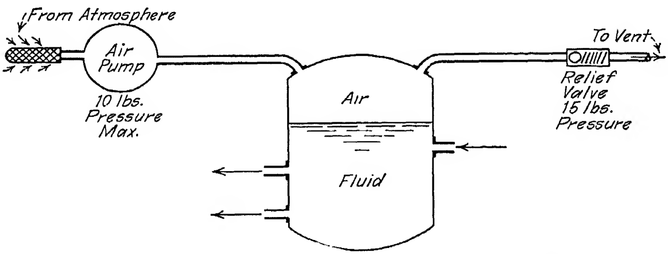


FIG. 51.—Schematic drawing of typical open-type supercharged reservoir.

It should be understood that not all these additions to the basic power system are required in all airplanes. However, most airplanes will be found to require one or more such additions,

depending on the particular conditions under which the airplane is expected to operate.

Previous chapters have covered additions to the basic hydraulic systems to bring both the subsystems and the power system up to date. The special requirements of the power system have now been described. There are also special requirements in each subsystem, such as landing gears, flaps, cowl flaps, etc.

CHAPTER V

SUBSYSTEMS

DESIGN REQUIREMENTS

Subsystems may be divided into two classes on the basis of their special design requirements. Thus, when a new subsystem is contemplated, its special requirements may be arrived at by reviewing the special requirements of other subsystems falling in the same classification.

Hydraulic systems are primarily used in aircraft when either high power or controllability is required. On this basis subsystems may be classified as follows: noncontrol systems, in which the primary purpose is to increase the pilot's power available; and control systems, in which the primary purpose is to provide smooth control either with or without higher powers than the pilot could deliver unaided.

Noncontrol systems can be divided into weight-loaded and air-loaded systems. The air-loaded systems include wing flaps, cowl flaps, and bomb doors, and the weight-loaded systems include such systems as landing-gear retracting systems. The chief difference is that accelerations encountered in flight affect weight-loaded systems but do not affect air-loaded systems.

Control systems can be divided into follow-up (position) controls and power-amplifying (load-feel) controls, position plus load-feel systems, and automatic systems, including systems in which a sequence of events is required.

Representative systems in each of these divisions will be taken up in turn, and the factors influencing the design of each system will be given. From these representative systems, it will be possible to design any subsystem by setting up its detail requirements. Only the hydraulic system from the valve through the operating cylinder will be considered, for it is obviously outside the scope of this book to cover the design of the parts being operated, *i.e.*, the landing gear, wing flaps, etc.

To design a subsystem or operating system and arrive at lines and units of the proper size and strength, it is necessary to know three things about the system, as follows: (1) loads, (2) speeds, and (3) special requirements.

The loads on the system are divided into operating and design loads. Operating loads, *i.e.*, those loads under which the system normally operates, determine the size of the operating cylinder or motor. Design loads, *i.e.*, the maximum loads that can ever be applied to the system, determine the burst strength of the lines and units, after the inclusion, of course, of proper factors of safety.

The speed of operation of the system is usually determined by its function and the desired operations of the airplane. The maximum required speed at the lowest required temperature determines line sizes and pump displacements, for this is the condition of maximum flow and maximum fluid resistance.

Special requirements are determined by the function of the system and the design of the airplane. They include such system requirements as approximately constant speed regardless of temperature or the inclusion of such auxiliary devices as landing-gear latches, wing-flap relief valves, or dashpots for high-speed units.

NONCONTROL WEIGHT-LOADED SUBSYSTEMS

A representative weight-loaded subsystem is the landing-gear retracting system. The normal loads, which determine the size of the operating struts, usually vary throughout the travel of the retracting strut; for the leverage of the weight changes as the gear is retracted, and the air loads also change. This varying load can best be determined by plotting a load-stroke diagram for the landing gear. This is usually done by drawing a number of diagrams of the landing gear in several intermediate positions between fully extended and fully retracted, and calculating the loads on the retracting strut for each of these positions. It is good practice to plot weight load, air load, friction load, door-operating load, bungee load, and any other loads independently and then draw an overall load curve; this procedure makes it possible to change weights, bungee strengths, etc., without complete recalculation. The size of the cylinder can now be determined by increasing the maximum load from this curve by 10 to 25 per cent to allow for line loss and then dividing by the pressure available (regulator

setting) to get the piston area. If the strut acts in tension, this resulting area will be the net area, and the rod area must be added to get the total cylinder area.

If the maximum load occurs at a peak in the curve and the other parts of the curve are well below the peak, then the line loss at the peak need not exceed 10 per cent. The line loss available for retraction will be greater than 10 per cent; for it is the difference between the pressure available and the average, rather than the peak, pressure required.

If the load curve is relatively flat-topped, *i.e.*, if the peak load is constant over 50 per cent or more of the retracting travel, then

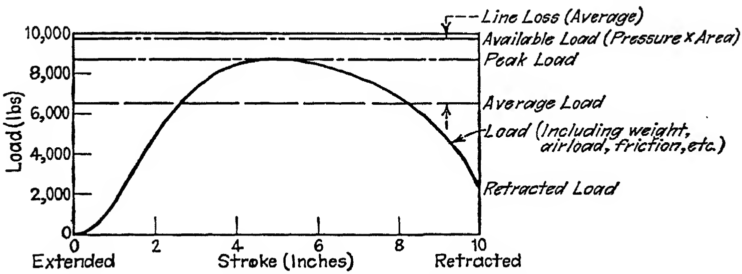


Fig. 52.—Typical load-stroke curve.

the peak line loss must be 25 per cent or more of the pressure available, for in this case the peak line loss will determine the retracting time; and if too small line loss is allowed, excessively large lines will be required. A typical load curve for a landing-gear strut is shown in Fig. 52.

The design loads, which determine the strength of the parts of the landing-gear retracting system, may result from any one of several conditions. The condition of maximum speed generally produces the highest air loads on the doors and their operating mechanisms. These loads, when combined with the loads caused by the weight of the landing gear and with bungee loads, if any, should be determined and translated into terms of pressure in the retracting strut.

Another design condition that often determines the design of retracting struts is the maximum accelerated-load condition. An accelerated load is the increased load that is applied to all members of the airplane when it is pulled out of a dive or flies through

a vertical gust. Maximum loads may occur in either of two conditions, as follows: While the gear is being retracted, the airplane is flown at lower than its maximum speed and is subjected to lower accelerations than the maximum. Accelerations in this condition normally do not exceed load factors of 2 to 3. When the gear is fully retracted, it is possible to get load factors as high as 6 to 7 on the landing gear. The proper load factor to use is the light-load load factor, which is used for the design of the structural parts of the airplane. This load factor is specified by the Civil Aeronautics Authority for commercial airplanes and by the Air Corps or the Navy for military airplanes.

For the condition in which the gear is being retracted, new curves of load on the gear should be drawn, including weight loads multiplied by the load factor, air loads, bungee loads, and any other loads that increase the loads in the retracting strut. Friction loads need not be considered; for the pressure in this condition normally is above that available, and the gear stops moving until the acceleration is reduced.

For the condition in which the gear is fully retracted, it is not necessary to draw curves, for the gear is in only one position. The weight of the gear should be multiplied by the proper load factor, and the air loads from doors, bungee loads, etc., should be applied. As in the previous case, these loads should be translated into terms of fluid pressure for the design of the hydraulic system.

Another condition that may affect the design of the hydraulic system is the pressure created by the expansion of fluid in the system with increase in temperature. This pressure is limited by the setting of the relief valves in the system.

The normal, or operating, pressure and each of these special design pressures must be multiplied by the proper factor of safety to get the required burst pressure for each condition. When the burst strength is calculated using the ultimate strength of the materials, the factors of safety should be as listed in the outline on page 89. The application of these loads and pressures to the detail design of units will be covered in Chap. IX, Hydraulic Unit Design Procedure.

The speed of operation of the landing-gear system varies with the type of airplane. On single-engine airplanes, it is necessary to get the gear up rapidly in order that the drag of the airplane will be reduced to a minimum as rapidly as possible so that the airplane

can climb over any obstacles that it may encounter immediately after take-off. On bimotored airplanes the retracting speed requirements are the strictest of those for any class of airplane because modern bimotored airplanes will usually continue to climb after failure of a single engine after the landing gear is retracted, whereas they will not climb if the landing gear is extended. This means that it is essential to the safety of a bimotored airplane that the landing gear be retracted as soon as possible after the take-off. At present, it is considered necessary to be able to retract the landing gear on a bimotored airplane in 10 sec. or less after the failure of one engine.

The size of the retracting strut having been determined from the normal retracting loads on it, the volume of fluid required to retract the landing gear is now known. The speed requirement combined with the volume of fluid required gives the flow required to retract the landing gear.

The two factors necessary to calculate the size of lines and valves required in the landing-gear system are now known, *i.e.*, the flow and the line loss, which is the difference between the average retracting pressure and the system pressure. Calculations for line loss must be run considering the lowest temperature at which it is necessary to obtain the required speeds. These temperatures are usually given in the airplane specification. They may range from -40 to $+32°$ F. For final calculations, it is permissible to take into account the fact that because it is coming from the circulating system the oil going into the struts is warmer than the oil returning from the struts. It is of course necessary to use the lowest temperature because that is the condition which gives the maximum fluid resistance.

It should be borne in mind that the pump capacity or the capacity of the pump plus accumulator must be sufficient, with one engine dead, to ensure proper retracting time.

The special requirements for the landing-gear retracting system may include one or more of the following items:

1. Landing gears very often incorporate latches to hold the gear in the up or the down position or both, after failure of the hydraulic retracting mechanism. These latches may be operated in any one of several ways. If they are mechanically operated, they are outside the scope of this volume and need not be considered further. If they are operated by an auxiliary cylinder, it is neces-

sary to consider the fact that the main cylinder may put loads on the landing gear which tend to bind the latch. The auxiliary cylinder must have sufficient power to unlatch the latches, overcoming any binding, which implies a high leverage between the auxiliary cylinder and the latch. If special valving is introduced so that the retracting fluid first goes into the latch cylinder and opens the latch before going into the main cylinder, then of course no binding can take place, for the main cylinder will be unloaded while the latch is operating. Usually, however, a simpler landing gear results when the latch is operated by the first portion of the travel of the main cylinder rather than by an auxiliary cylinder. In this case, it is also necessary to consider possible binding, which may result because of the fact that the pull of the main cylinder on the latch usually also tends to retract the gear and therefore may cause binding.

2. In modern high-performance airplanes, doors are ordinarily used to close completely the aperture into which the landing gear retracts. These doors are usually operated by a mechanical linkage from the landing gear and therefore have no effect on the hydraulic system except to add their loads to the other loads from the landing gear. Occasionally it is necessary to provide separate door-operating cylinders. It then becomes necessary to ensure that there is no interference between the doors and the landing gear. This may be done in one of two ways, as follows: by means of hydraulic connections such as valves or flow-splitting devices, which tend to divert most or all of the flow to the door cylinders first, then to the landing gear; or by means of series valving, in which operation of the landing-gear valve directs the fluid into the doors, these, when open, operating another valve, which directs the fluid to the landing-gear retracting struts. When the main operating valve is reversed, the landing gear operates first and in turn operates the door valve, which then closes the doors.

Both these systems have disadvantages. Series valving has the disadvantage that, if the doors jam, advantage cannot be taken of the high-powered retracting strut to force the gear down. The hydraulic interconnecting system has the disadvantage that under unusual conditions, such as partial extension followed by retraction, interference may occur between the doors and the gear.

3. Where the main landing-gear valve is remotely operated, it is often specified that a valve be added that can be directly oper-

ated by the pilot to let the fluid out of the landing-gear retracting lines so that the gear can be lowered in case of failure of the valve-operating mechanism or valve, which would otherwise trap fluid in the retracting lines and prevent the gear from being lowered. This emergency extending valve is also useful as a safety device because when it is in the open position the landing gear cannot be retracted on the ground by inadvertent operation of the main landing-gear valve.

4. When the landing gear is lowered by means of an emergency extending device, or on the ground with the airplane on wing stands, the weight of the landing gear often forces fluid out the retracting line faster than atmospheric pressure can force fluid from the reservoir through the four-way valve into the extending side of the operating cylinder. This results in more fluid being returned to the reservoir than is being drawn from it, which, unless excess capacity is provided, may result in overflowing the reservoir. To prevent such overflow, valves may be installed to permit flow from suction lines or other large reservoir lines adjacent to the landing gear into the landing-gear extending line, to provide a low-resistance path for fluid into the extending side of the landing-gear cylinder.

5. With high-speed retraction of heavy landing gears the shock loads built up in the retracting system when the landing gear reaches the end of its travel may be considerable. To prevent damage to the retracting mechanism or adjacent structure, it is often necessary to provide dashpots or shock-absorbing mechanisms at the ends of the travel of the retracting strut so that the moving weights may be decelerated gradually rather than abruptly. The design of these dashpots will be covered under the design of operating cylinders.

6. On nose wheels or tail wheels, which are free to swivel, some device must be incorporated to prevent retraction in a swiveled position, for this might cause damage to the exterior of the airplane adjacent to the well into which they retract. When this device takes the form of an automatic centering device, it does not affect the hydraulic system. Otherwise, it is necessary to have some sort of hydraulic centering device or a lock to prevent retraction, *i.e.*, to shut off fluid in the retracting lines, when the gear is not centered.

The normal and design loads, speeds, and special requirements of landing-gear retracting systems have now been described. Any

weight-loaded operating system in the airplane is likely to have approximately the same requirements. The same factors will normally have to be considered in its design.

NONCONTROL AIR-LOADED SUBSYSTEMS

Typical air-loaded systems, *i.e.*, systems in which the principal load on the system is an air load rather than a weight load, are wing flaps, engine cowl flaps, and bomb doors.

Operating loads on wing flaps are simply those caused by the air loads at the maximum air speed at which it is required to extend the wing flaps fully. Unless there is a very similar airplane already operating from which the loads can be determined, it is necessary to determine the loads by wind-tunnel model tests or by calculation. In either case, ample margin, probably at least 50 per cent, should be provided to allow for errors in testing and calculation. In the case of unconventional flap designs, particularly when the loads are calculated, even higher margins are desirable.

Design loads are obtained by multiplying the operating loads by the proper factor, which will normally be about 2.5 (see outline, page 89). The only other design load is the pressure resulting from temperature expansion; this is determined by the setting of the relief valves in the system.

Wing-flap operating speeds are usually slow. However, the slow speeds cannot be taken full advantage of when the size of the lines is determined; for, because these speeds affect the operation of the airplane, they must be kept somewhere near constant regardless of temperature, which usually means that oversize lines are required, together with orifices to get the proper speed. Also, in the original design of the airplane it is impossible to calculate the speed with sufficient accuracy, so that it is necessary to provide oversize lines and orifices that can be varied on the first airplane to get exactly the most desirable operating time. It is desirable to retract the flaps slowly to prevent stalling the airplane. This is because the stalling speed of the airplane with flaps extended is lower than the stalling speed with flaps retracted. The airplane must be allowed time to accelerate while the flaps are being retracted. On the other hand, it is desirable to retract the flaps rapidly to reduce the lift once the airplane is on the ground in order to make the brakes more effective and, by reducing the lift, to

make the airplane less likely to be overturned by gusts. It is also desirable to retract the flaps rapidly to prevent damage from water thrown up by the wheels after landing on wet or muddy fields. The final retracting time chosen is usually based on a compromise between these factors and may be 5 to 15 sec. Extension time of the flaps is usually not critical and may be between 3 and 15 sec.

There are certain special requirements that apply to wing-flap systems on most airplanes. A relief valve is usually incorporated in the extending lines, which can be set to allow the flaps to retract of their own accord when the air loads on them exceed the safe strength of the flaps. This cannot be a conventional relief valve because it must be set at a pressure below that of the main system. The design of this type of valve will be discussed in Chap. IX, Hydraulic Unit Design Procedure. It generally incorporates some sort of three-way valve arrangement operated by the pressure in the extending line so that when this pressure rises above a set value it first shuts off the pressure supply line, then opens the cylinder to the reservoir. When such a valve is incorporated in the system, it is also desirable to put a check valve from the reservoir line to the flap-up line. Thus, when the flaps are blown up by excessive air load, oil can be drawn into the retracting side of the flap strut from the reservoir line and so will not overflow the reservoir or possibly draw air into the flap struts past the piston-rod packing.

Another special requirement that is peculiar to wing-flap systems is the necessity for accurate synchronization of movement of the flaps on left and right sides of the airplane to eliminate a tendency of the airplane to roll. It is customary in small airplanes, if the flaps are hydraulically operated, to actuate them by a single cylinder operating the left and right flaps by mechanical means, which, of course, ensures synchronization. In larger airplanes the weight of this type of system becomes prohibitive, and the lighter hydraulic synchronization must be resorted to. Where the load on the flap-operating strut continually increases from the flap-up to the flap-down position, the loads themselves will tend to synchronize the flaps; for if one flap should get slightly ahead of the other flap, its load would increase and therefore the fluid would flow into the other flap strut until the loads were again balanced. This air-load synchronization is possible only where the frictional forces are relatively low. Obviously, if there were high friction on one side of the airplane and low friction on the other, the flaps having

the lower friction would extend farther for the same pressure and the airplane would tend to roll. Air-load synchronization is also not practical where the load curve for the flap struts decreases at any time during the extension of the flaps; for in such a case there will be two points at different extensions at which the flaps can be in equilibrium, so that one flap could be extended farther than another and still require the same pressure, though its rolling moment on the airplane would almost certainly be different.

Hydraulic synchronization of flaps can be accomplished by means of flap struts in series. This, however, has the disadvantages that were previously considered with reference to series systems and, as was found at that time, is uneconomical.

It is sometimes considered desirable, when the flaps are operated separately, *i.e.*, without mechanical interconnection, to have a separate indicator on each flap. Advantage can be taken of these indicators to synchronize flap operation. The indicators are connected to valves in the flap lines in such a way that the flow to the flap which is leading is restricted, thus slowing it down and giving the other flap a chance to "catch up." Synchronization can also be accomplished by means of some sort of flow-splitting device, which usually takes the form of two gear pumps, one in each flap line, the shafts of which are mechanically interconnected so that the pumps are forced to rotate at the same speed, an equal division of flow between flaps being thus ensured.

Another air-loaded system in common use is the engine cowl-flap system. The normal loads, which determine the size of the operating cylinders, are those required (1) to hold the cowl flaps open at the climbing speed of the airplane, when the engine power is high and the air speed, and therefore the cooling, are low; and (2) to hold the flaps closed against the maximum air speed at which the airplane can be flown. In this condition, friction need not be considered because it is not necessary to close the flaps at maximum air speed but only to hold them closed in this condition.

Loads on cowl flaps cannot be accurately determined in advance of flight test; thus, it is usually necessary on a new cowl-flap installation to allow a considerable margin of safety for possible increase in load over the calculated loads. Wind-tunnel tests are fairly accurate if made on an actual nacelle with an actual engine and with the wing in its proper position with respect to the nacelle. Such tests, however, are rarely made.

Maximum loads for the design of the cylinder are those required to hold the cowl flaps closed at the maximum diving speed of the airplane or to hold the cowl flaps open up to the maximum speed the airplane can attain with flaps open, if a relief valve is not provided to allow air loads to close the cowl flaps. Also, of course, the effect of temperature expansion must be considered.

Cowl-flap operating speeds are usually slow because most present-day cowl flaps are set to intermediate positions by operating the valve, waiting until the cowl flaps arrive at the desired position by watching either the flaps themselves or an indicator, then shutting the valve off again. This requires an operating time of not less than 5 sec. so that an intermediate position can be chosen with some degree of accuracy. The operating time should not exceed 10 sec. so as not to take the pilot from his other duties for too long an interval. Because these times should vary little with temperature, it is necessary to use larger lines than would otherwise be required, and incorporate an orifice in the system. The lines still will be very small because the powers required to operate cowl flaps are low.

To reduce the amount of attention the pilot has to devote to his cowl flaps, it is possible in some airplanes to compromise and provide only three cowl-flap positions, open, closed, and trail, which is used in climbing or high-speed flight in hot weather. In the trail position, both cylinder ports are open to the reservoir so that the cowl flaps are free to assume any position to which the airloads tend to force them.

Because of the necessity for increased performance, future airplanes will probably use some sort of follow-up or position control in which the position of the cowl-flap valve handle or lever corresponds to the position of the cowl flaps. With this system, when the cowl-flap lever is put to the three-quarters open position, for example, the cowl flaps travel to that position and remain there. The method of doing this hydraulically is taken up later in the chapter, in the section on Control Subsystems.

A still further refinement of cowl-flap control that can be expected in the future is the automatic control, in which the cowl flaps automatically take the position necessary to keep the engine at the proper temperature. This also will be described in the section on Control Subsystems.

The third typical air-loaded system to be discussed is the bomb-door system. Because the bomb doors have to be operated at high speeds, their normal operating loads are those required to operate when the airplane is at high speed. Bomb-door loads cannot be calculated with any degree of accuracy but can be determined with fair accuracy from wind-tunnel models. With large airplanes, this should be done to reduce the weight that inevitably results from using oversize struts to allow for possible unknown loads.

When bungees are incorporated to ensure emergency door opening in case of hydraulic-system failure, the hydraulic system must also have sufficient power to close the door against the bungee loads. The design loads are those which come from the highest air loads on the bomb doors. These are ordinarily the air loads prevailing in a dive plus the bungee loads if any and, if the doors are heavy, plus accelerated weight loads.

There is no very definite speed requirement in the operation of bomb doors. Ordinarily, operating speeds range from 2 to 8 sec.

One special requirement of bomb doors that has not as yet received much attention is the requirement that they operate at high altitudes, which of course does not apply to any of the systems previously described. This sometimes involves some sort of heating arrangement for the bomb-door operating cylinder and lines. This may be through the ship's ordinary heating facilities if it has any, or it may be done hydraulically by putting an orifice in the piston, which allows a constant circulation through the bomb-door strut thus keeping the temperature up above the freezing point of the oil, which, with Sperry oil, is about $-50°F$. Something of this sort is necessary on high-altitude bombers.

Another special requirement of the bomb-door system is that it must be possible to open the doors in event of failure of the hydraulic system. This emergency opening system may consist of a bungee or of a mechanical system such as a crank-and-cable system.

The three systems last described are typical of air-loaded systems, of which the primary function is to operate some unit against air loads. Other air-loaded systems, such as surface-control systems, are used in the airplane; but their operation is primarily one for which accurate control is required, and therefore they properly fall into the class of systems to be discussed under control subsystems.

Any air-loaded power system can be designed by reviewing the requirements for the three systems just described and incorporating the necessary special requirements.

CONTROL SUBSYSTEMS

The subsystems covered earlier in the chapter were those in which the reason for using hydraulics, rather than some other form of power, was because of the high powers that can be developed by such systems when compared with other forms of power transmission. The systems to be covered in this section are those in which hydraulics are used because of the controllability thus obtained rather than simply the power. Such systems can be roughly divided into four major classes, as follows:

Power-amplifier systems are those in which it is desired to multiply the operator's force by some given factor, the output force being kept always proportional to the input force. These are sometimes called "load-feel" systems because the operator can feel the load against which he is operating, though in a reduced amount.

Position follow-up systems are systems in which it is desired to make a movement at a distance duplicate the movement of the operator's control and in which a cable or push-rod system cannot be used because of lost motion, difficulty of installation, or high power required.

Position and load feel are sometimes combined, as in power surface control systems, in which the pilot can feel both the position of the control surfaces and the loads on them, though the loads may be reduced in any desired ratio.

Automatic controls, which include all controls that operate without attention on the part of the pilot, such as thermostatically controlled oil radiator shutters, are not in common use at present but are coming into use as fast as the need for them arises.

The first of these control systems, the power-amplifier, or load-feel, system, came into use first in power-brake systems, in which it is important that the feel of the pedals be similar to that of manual brakes, because the degree of brake application is judged to a large extent by the feel of the load on the toe pedals. The essential feature of a load-feel system is that the pressure in the system is always proportional to the load on the control, which in the case of brakes is the brake pedal. This is similar to the load

feel of a manual-brake installation, the advantage of the power system being that the pilot can control much more power than he can apply directly, the control for large airplanes thus being the same as for small airplanes. The limiting size for direct operation of brakes is reached in airplanes of about 15,000 lb. gross weight.

The device that produces this pressure proportional to load is usually called a *brake valve* and incorporates a pressure inlet valve, a return, or outlet, valve, and a piston acted upon by the pressure in the brake line. Such a valve is shown in Fig. 53.

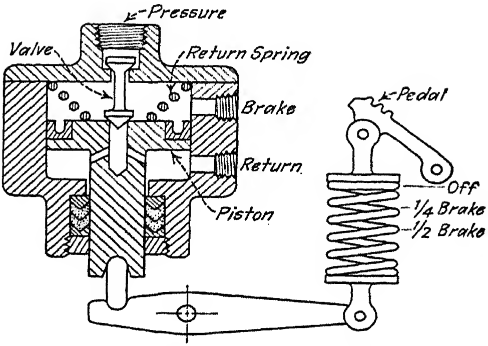


FIG. 53.—Schematic drawing of one type of brake control valve.

The operation of this valve is as follows: When the pedal is depressed to the half-pressure position, the piston moves up, carrying the valve with it, and opening the pressure port. Fluid flows in through the pressure port and out to the brakes until the brake pressure rises to just above half pressure, when the piston moves down, compressing the spring slightly and shutting off the pressure port. The load on the pedal is now half the maximum pedal load, and the pressure is half the maximum brake pressure. When the brakes are to be released to the quarter brake position, the pedal is released to the new position, thus momentarily decreasing the spring load. The piston then moves down under the influence of the now unbalanced brake pressure, causing the return port in the piston to open and allowing fluid to run out and brake pressure to drop until it reaches a value just below the spring load, when the spring extends slightly and closes the return port. The valve is now in equilibrium in the quarter brake position.

Proportionality is ensured by the fact that the load felt on the pedal is due solely to the pressure on the top of the piston, which is, of course, the brake pressure. It should be noted that the spring does not affect the load on the toe pedal, only the movement of the toe pedal. If the spring is omitted, the toe-pedal travel is too short to feel natural and the kickback when the brake pressure comes up is more noticeable.

The normal loads on the system are those produced when the pedal is depressed to the limit of the pilot's ability or when the stop is reached, if one is incorporated. These loads are also the design loads, when multiplied by the proper factor of safety (see page 89). The effect of temperature expansion in the brake line is simply to back off the piston and relieve the pressure.

As in all systems that simulate manual operation, the speed of operation must be sufficiently rapid so that no lag is noticeable. This involves a speed of operation of about 1/4 sec. from zero to full brake. Because the displacement of brake operating cylinders is usually small, the flow is relatively low in spite of the high speed required.

This system is a load-feel system because the pilot's feel is proportional to the brake pressure, although the position of the toe pedal is not proportional to brake position. Actually, in the systems just described, the pedal position is proportional to the brake pressure. Any load-feel system is likely to follow the same sort of design, with special requirements to fit the particular system being designed.

The second type of control system is one in which the position of the control is always relative to the position of the unit being operated. This type of system is used for such purposes as wing-flap, cowl-flap, or radiator-shutter control and is ordinarily used in preference to a manual control because of a high power requirement. It is used in place of a straightforward hydraulic system in order to get more accurate positioning in less time than by the procedure of watching an indicator and operating a valve, as is required with a simple hydraulic system.

The synchronization of the cockpit control with the unit being operated can be accomplished either mechanically or hydraulically. Where the system is primarily used to develop high power, as in wing-flap systems, mechanical synchronization is sometimes used. Where the horsepower required is low, as in cowl-flap or radiator-

shutter systems, the complication of a mechanical follow-up system is hardly warranted; for if it is necessary to put in cables for synchronization, the unit might almost as well be mechanically operated.

Hydraulic synchronization involves the use of two cylinders in series. This requires synchronizing valves at the ends of the travel to allow leakage to be made up and to correct for temperature expansion. A hydraulically synchronized remote position control system is shown in Fig. 54.

The system shown in Fig. 54 operates as follows: When the cockpit control handle is pushed to the left, the synchronizing, or

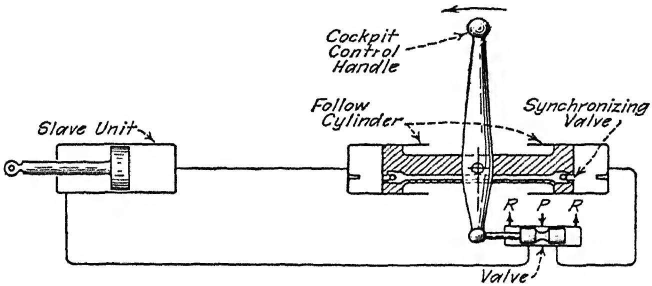


FIG. 54.—Schematic diagram of hydraulically synchronized remote position control system.

follow-up, cylinder acts as a pivot and the valve operates. The valve in this case will operate to direct fluid into the right-hand end of the cylinder, thus moving the piston to the left and tending to return the valve to neutral. If the handle is kept pushed to the left, oil will continue to flow into the right side of the cylinder, moving the piston to the left and forcing the oil out the left side of the cylinder into the operating, or "slave," unit. Oil coming out the other side of the slave unit returns through the four-way valve to the reservoir. When the handle is stopped, oil continues to flow into the follow-up cylinder, moving the piston sufficiently far to the left to shut the valve off.

Because of the fluid trapped between the left end of the follow-up cylinder and the right end of the slave cylinder, it is obvious that these two units will always be synchronized except for leakage and temperature changes. When the unit reaches the left end of its travel, the left synchronizing valve in the piston is forced open

by the pin in the end of the cylinder and oil flows through the passage in the follow-up piston into the slave cylinder, forcing it to the extreme left end of its travel. The system is therefore synchronized at the left end of its travel. When the operating handle is at the extreme right-hand end of its travel, the right synchronizing valve opens and the oil comes through the piston. The oil coming through the valve into the left end of the slave cylinder forces it to the extreme right. The oil coming out the right end of the slave cylinder can go through the open synchronizing valve, through the four-way valve, and back to the reservoir. The system is now synchronized at the right-hand end of its travel.

The lost motion at the end of the handle required to operate the valve can be kept negligibly small by the use of poppet valves. This system can also be used as a spring return system by substituting a spring at the left end of the slave cylinder; in this case, only a three-way valve, rather than a four-way valve, is required. Such a system is satisfactory for easily controlled units such as engine controls.

The design loads on the units are those produced by some arbitrary high load on the end of the handle, in combination with the maximum pressure. A reasonable design load for the end of a handle that is readily accessible is 180 lb.

Position control systems do not have to operate as fast as load-feel systems. Low-powered systems such as oil radiator-shutter or engine control systems should operate through their entire travel in 1 to 3 sec. High-powered systems such as wing flaps may be operated satisfactorily at 1 to 5 sec.

The third type of control system is a system combining both position and load-feel. Such systems are necessary where a manual control system must be duplicated as accurately as possible. For instance, in surface control systems in large airplanes, where the pilot is unable directly to apply sufficient force comfortably to control the airplane, it is necessary to provide a system that will multiply the pilot's power in some fixed ratio while retaining the full feel of the loads on the airplane and an indication of the position of the controlled surfaces. As in position control systems, the follow-up mechanism can be mechanical or hydraulic. Surface-control-system reliability is of vital importance to the safety of the airplane, and so far such systems have been made with mechan-

ical follow-up systems that not only can position the surface more accurately but can act to transmit power, at least to the extent of the pilot's ability to apply it, in case of failure of the hydraulic system. A typical surface control system is shown in Fig. 55.

The operation of this system is as follows: When the pilot pulls on the elevator control column, *i.e.*, moves it to the right, the cable control arm, which is pivoted on the same axis as the control surface, rotates to the left, moving the valve piston and directing oil into the left end of the booster cylinder. This forces the cylinder to the left, thus moving the control arm and surface. This

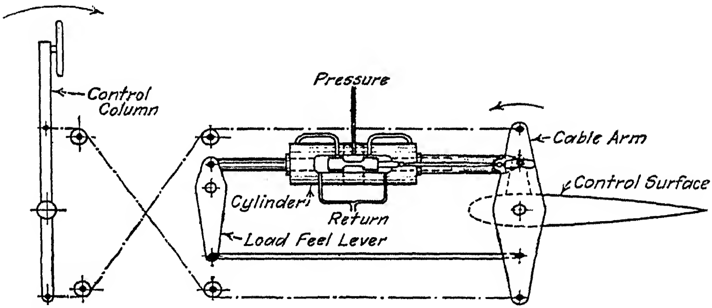


FIG. 55.—Schematic drawing of hydraulic surface control booster system using mechanical follow up.

action continues until the valve body, which is mounted on the cylinder, catches up with the valve piston and shuts off the flow of fluid. Thus the control surface always follows the movement of the cable control arm. If the end of the booster-cylinder piston is rigidly connected to the structure, there is no load feel and the system is irreversible. However, if a linkage is incorporated in the system to connect the stationary end of the booster cylinder to the cable control arm through a lever, then part of the load on the surface, which is equal to the load in the cylinder, will be fed back to the cable control arm where it can be felt by the pilot. The percentage of the cylinder (surface) load that will be felt by the pilot is directly proportional to the leverage ratio of the lever connected to the cylinder end. If the long end of the lever, which is the end connected to the cable control arm, is four times the length of the short end, which is the end connected to the cylinder, then the pilot will feel the load on the surface, reduced to one-fourth its actual value.

The only difference in the hydraulic parts of the booster system caused by the addition of the load-feel linkage is the fact that the cylinder stroke is shortened slightly because the left end of the cylinder moves in the same direction as the right end, though at a slower rate.

Load feel can also be provided by anchoring the end of the booster-cylinder piston, but connecting the cylinder through pipes to a cylinder in the pilot's control cable. Thus the cylinder (surface) load, in this case in the form of hydraulic pressure, will be transmitted to the pilot in the ratio of the area of the load-feel cylinder in the pilot's control cable to the area of the booster cylinder. (The leverages of each cylinder must also be taken into account.)

Because in the position and load-feel type of system the attempt is to duplicate as closely as possible a mechanical system, the loads, both normal and design, will be those which would be applied to a mechanical system, except that the effect of pressure and of temperature expansion must be considered.

Speeds of operation must be as rapid as possible, for with the mechanical system the speed of operation is limited only by the force that can be applied and the inertia of the surface being moved. The usual requirement for surface control systems is 1 sec. from neutral to hard-over in either direction.

In some types of control system, in fact, in most types, more than one of these three types can be used. For example, all three types have been used for nose-wheel steering on airplanes equipped with tricycle landing gears. The use of the first type of system—the load-feel system—results in a system in which the movement of the nose-wheel steering control, which may be a separate control wheel, the aileron control wheel, or the rudder pedals, is proportional to the force tending to turn the nose wheel. This means, for instance, that for half travel of the control, the nose wheel may turn almost the limit of its travel at low speeds and only a few degrees at high speeds, where the force required to turn it is much greater. This results in an equal centrifugal force on the airplane for equal displacements of the control wheel and produces a control that gives full control at low speeds and is not too sensitive at high speeds.

When the second type of control is used, the position control, with no load feel, an irreversible control system results, in which

the position of the nose wheel is determined by the position of the cockpit control. This results in a sensitive control at high speed; but because the centrifugal force, or side load, on the airplane can be readily felt by the pilot, such a system is controllable. It has advantages in parking the airplane or starting from a standstill, in that the pilot knows the position of the nose wheel.

When the third system, combined position and load feel, is used for nose-wheel steering, a system results that is practically a duplicate of the steering system in an automobile. That is, both the loads and positions are felt on the cockpit control. Like an automobile, this system is sensitive at high speeds; but because the loads are high, overcontrol is prevented.

Surface control systems have been built using each of these three types, but the third is the only one that has proved entirely satisfactory in practice. In theory, at least, the first should also prove satisfactory and should reduce the sensitivity of control at high speeds. However, mechanical connection to take care of possible hydraulic failure is difficult to arrange with this system, which limits its use, at least until hydraulic systems attain a higher degree of reliability.

Automatic controls have not, as yet, received the attention they deserve. Automatic systems can be arranged sometimes with less complication than a conventional hydraulic system to operate some controls that now require attention on the part of the pilot. For bimotored single-place airplanes, this problem of pilot attention is serious, and there is a wide field for further application of automatic controls.

Some of the controls to which automatic controls could be applied are as follows: oil radiator shutters, which can be controlled by the temperature or the viscosity of the lubricating oil; cooling radiator shutters, which can be controlled by the temperature of the cooling liquid; and engine cowl flaps, which can be controlled by engine-head temperatures.

Automatic control systems include three elements. The first of these is an element responsive to the effect that it is desired to control. In the case of cowl flaps, this would take the form of a bimetal or liquid-filled element responsive to the engine temperature. The second element is a valve operated by the control element, and the third is the operating cylinder. One precaution that it is necessary to observe in the design of such systems is the

provision of "droop," *i.e.*, sufficient range of control to avoid hunting. Dead-beat systems—*i.e.*, systems that attempt to control temperature, viscosity, etc., to one point rather than over a range —often run into hunting difficulties. A typical automatic system, in this case an engine cowl-flap system, is shown in Fig. 56.

Consider the operation of the automatic cowl-flap system shown in Fig. 56 first without the addition of the follow-up rod, *i.e.*, with the valve body fixed. In this case, if the temperature rises above the neutral, or desired, position, the bimetal element moves to the right. This moves the four-way valve and directs fluid to the left-hand end of the operating cylinder, which opens the cowl flap.

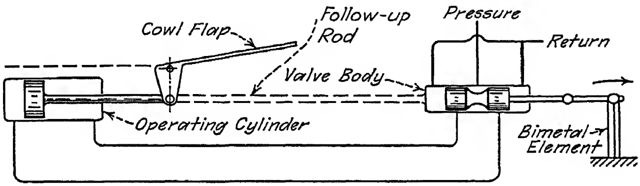


Fig. 56.—Automatic engine cowl-flap control system.

However, because the cowl flaps open relatively rapidly—at least, more rapidly than the engine temperature changes—the cowl flap will continue to open until it is full open before the engine will have cooled down sufficiently to bring the bimetal element back to neutral. The wide-open cowl flaps will induce such a large flow of air that the bimetal element will go past neutral in the cold direction, whereupon the cowl flaps will immediately close all the way. Thus, hunting results from the use of this type system, which tries to keep the engine at one fixed temperature corresponding to the neutral position of the bimetal element.

The magnitude of the hunting can be reduced by slowing down the operation of the cowl flaps, but not always to the point where satisfactory operation is attained. The addition of a follow-up rod, shown dotted between the cowl flaps and the body of the valve, changes the operation so that the cowl-flap system including this rod operates as follows: When the head temperatures rise so that the bimetal element moves to the right, the valve moves to the right, which directs fluid into the left end of the cylinder. This opens the cowl flaps and moves the valve to the right, catching up with the piston and shutting off the flow to the cowl flap. The

cowl flap and the bimetal are now in equilibrium in a new position, with the cowl flaps partly open and the head temperature slightly higher. Obviously, when the cowl flaps are wide open the head temperature will be above that corresponding to the neutral position of the bimetal element, and when the cowl flaps are completely closed the head temperature will be below neutral. This difference in temperatures over the range of cowl-flap operation is known as the "droop," and a system incorporating it is called a system having a "drooping" characteristic. Such systems are inherently nonhunting and must be used wherever rapid operation without hunting is required. The droop necessary to prevent hunting must be determined by experiment, for it depends upon the friction and other characteristics of the system.

The advantages of the application of such automatic controls to any control that must be operated by the pilot to keep a temperature, a viscosity, or a flow constant are obvious. Such controls have already been applied to propellers to keep engine revolutions constant and to supercharger regulators to keep manifold pressures constant.

These control systems are being used in a limited way by airplane manufacturers, and their further application is limited principally by a lack of knowledge of the design of such systems.

By deciding into what class any control system should be put, then following the general outlines given for the typical systems considered, it should be possible to design any type of control system. The general requirements of all hydraulic systems the airplane designer is likely to encounter having been described, the detail design of one of these systems will be discussed in the next chapter.

CHAPTER VI

DESIGN PROCEDURE

In previous chapters power systems and various types of subsystems in use in modern airplanes have been discussed, and the loads and speed requirements and any special design requirements in these systems have been given. This chapter will cover the design procedure for a typical aircraft hydraulic system. The final result of the design of a system is a schematic diagram showing the hookup of the various units and giving the sizes of the units and the connecting lines and a complete schedule of pressures, including relief pressures, operating pressures, etc.

The detail design of the units will be taken up later. For the design of the system, a complete description of the hydraulics of the system is needed. The design procedure for an aircraft hydraulic system proceeds by logical steps from the known factors to the final result. The first step in the design of a system is the determination of the operating loads on all operating cylinders. This cannot be done until the landing gear, wing flaps, and other parts of the airplane that are to be operated by the hydraulic system have been designed, at least in a preliminary manner.

The loads on the wing flaps, landing gear, etc., should be put into the form of curves of load required plotted against travel of the operating cylinder. The maximum load that must be operated in any condition can be taken from this graph. This maximum load should be increased by a factor varying from 10 per cent in the case of a load curve that is sharply peaked, where the average is considerably below the maximum, to 25 per cent in the case of a load that is nearly constant over the travel of the operating cylinder, to allow for flow resistance in the operating system. This is on the assumption, of course, that the loads on the graph include all loads up to the operating strut, which include weight loads, air loads, friction loads, and allowances, or margins, for possible increase in weight and air load over those estimated in the original design of the airplane.

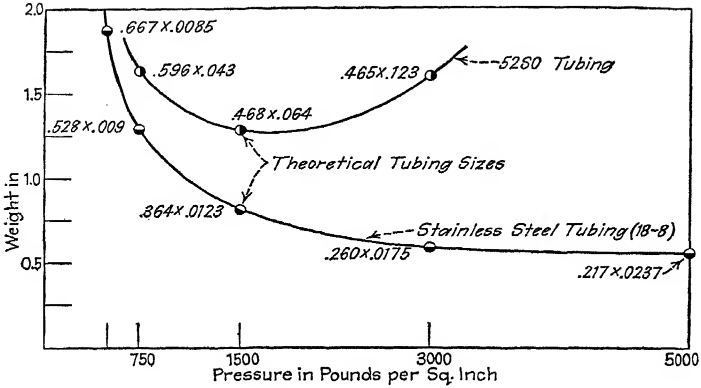


FIG. 57.—Weight of 10 ft. of 52SO aluminum-alloy and stainless-steel tubing and oil to transmit 10 hp. at 90 per cent efficiency and a stress of one-sixth the ultimate

$$S_{T\,\max} \quad \frac{P_1(D_0^2 + D_1^2)}{D_0^2 - D_1^2}$$

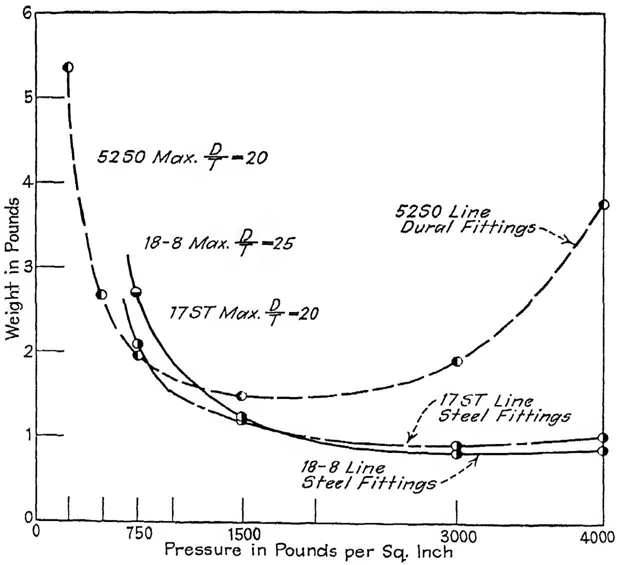


FIG. 58.—Weight of 10 ft. of tubing and oil to transmit 10 hp. at 90 per cent efficiency and a stress of one-sixth the ultimate, using tubing in the range of available D/T (including fittings spaced at 40 in.).

The nominal system pressure, *i.e.*, the cutout pressure of the pressure-regulating device, should be made as high as possible, at least up to 3000 lb./sq. in., in the interests of light weight. However, the limited availability of tubing, flexible hose, and packings suitable for these pressures and the lack of service experience as well as limited availability of units such as valves and pumps have

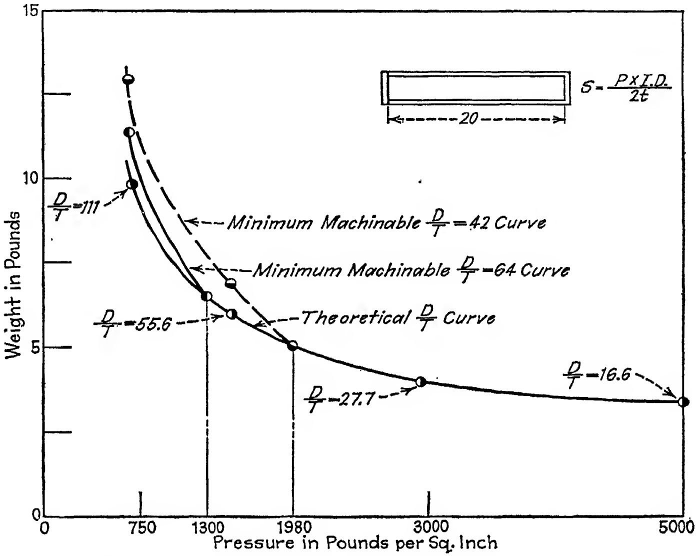


FIG. 59.—Weight of cylinder and oil to develop 50,000 ft.-lb. at a stress of 41,700 lb./sq. in. (⅓ of 125,000 lb./sq. in.).

caused most systems, particularly on small airplanes, to be limited to pressures of about 1000 lb./sq. in. The curves of Figs. 57, 58, and 59 show graphically the effect of pressure on the weight of hydraulic lines and hydraulic operating cylinders.

The cylinder size can now be determined by taking the maximum cylinder load, increased 10 to 25 per cent to allow for line loss, and dividing it by the pressure available in the hydraulic system at the time that that particular unit is to be operated, which normally is the nominal system pressure. This calculation gives the area of the cylinder. If the cylinder is of the type having a piston rod extending out one end of the cylinder, as most cylinders are, the

full area of the cylinder is available only when the cylinder is extending. When the cylinder is retracting, the area that has just been calculated is the annular area outside the piston rod and inside the cylinder bore. The stroke of the cylinder is, of course, determined by the mechanism that it operates.

Calculating for an airplane similar to the DC-3, the cylinder loads and sizes would be determined for the landing gear, wing flaps, cowl flaps, and brake systems as outlined above. After finding the cylinder sizes, the speeds of operation must next be determined. The areas and speeds being known, the maximum flow in the system can be calculated, and this will determine the size and power required and, at the same time, give a general idea of the size of the system. On the DC-3 no two subsystems are operated at the same time. Therefore, in order to determine the maximum flow, the flow for each of the subsystems will be determined in order to find the largest. The wing-flap operating cylinder is assumed to have an area of 5 sq. in. and a 10-in. stroke. The cowl flap and brake cylinders are sufficiently small so that, although they must be calculated, they will not determine the maximum flow in the system.

The required speed of operation of the landing gear is 10 sec. for retraction. There is no extension-speed requirement. The wing flaps must operate in 7 sec. Since the volume of the retracting side of the landing gear is $3\frac{1}{2}$ sq. in. (area), times 24 in. (stroke), times two cylinders, the volume that must be pumped in 10 sec. is 168 cu. in., and the average maximum flow required in the landing-gear system is 6×168, or 1008 cu. in./min.

The flow in the wing flaps will be 5 sq. in. $\times$ 10 in. stroke, or 50 cu. in. in 7 sec., making a flow of $50 \times {}^{60}\!/_{7}$, or 430 cu. in./min. Since the flow required by the landing gear is required at take-off speed, which corresponds to 3600 pump r.p.m., this requires a pump displacement of 0.28 cu. in. per revolution. Since the flow required by the flaps is required during approach, when the pump is running at about 1600 r.p.m., this requires a pump displacement of 0.268 cu. in. per revolution. Since it thus appears that the landing gear requires the maximum flow, it can now be said that the pump must deliver 1008 cu. in./min., or about $4\frac{1}{2}$ g.p.m., at 3600 r.p.m. Since the brakes are power brakes and therefore there is an accumulator, it will be assumed that the pump need only pump the average retracting flow.

Since it has been determined that the system is a 4½-g.p.m. system, which at 800 lb./sq. in., develops roughly 2 hp., the system of pump regulation to be used can now be decided upon. Since the horsepower is sufficiently high so that there would be an excessive amount of heating in the system if all the horsepower were to be turned into heat by forcing the fluid always through a relief valve as in a constant-pressure type of pump-regulating system, this system cannot be used. Since the horsepower is not sufficiently high to cause an excessive amount of hammering in the system and variable-displacement pumps or other shockless forms of pressure regulation are thus not required in this particular case, an intermittent form of pressure regulation can be used in which, when the pressure is between certain limits, the pumps circulate oil freely under no pressure. Another factor that determines the system of pump regulation to be used is the fact that in the DC-3 all the systems that are to be operated operate intermittently. Where there is a continuous drain on the system as, for instance, from the operation of power surface-control boosters, which are operated continuously in flight, the intermittent system of pump regulation is not satisfactory, because the regulator has to operate at a frequency of at least several times a minute, putting shock loads on the pumps, regulator, and lines throughout the system.

It is now possible to proceed to the design of the power system, for its horsepower has been determined and its major feature, the system of pump regulation, decided upon. The first step in the design of the power system is the determination of special units that may be required and the special design requirements for the system. In previous chapters, these special design requirements have been listed.

On the DC-3, because of the use of power brakes, an accumulator was used. Its capacity was determined by two criteria, as follows: (1) It had to be large enough to provide a sufficient quantity of oil under pressure to operate the brakes several times. (2) Because of the need to retract the landing gears in a short time, even with one engine inoperative, it was decided to make the accumulator sufficiently large so that, in the event of failure during take-off of the engine on which the hydraulic pump was mounted, the accumulator capacity would be capable of retract-

ing the landing gear, at least to the point where the drag of the airplane would be reduced to a reasonable minimum, which, in the DC-3, requires almost complete retraction. The accumulator actually used has a capacity of 250 cu. in. In using up these 250 cu. in. the accumulator drops from a system pressure of 800 lb./sq. in. down to its initial inflation pressure of about 300 lb./sq. in.

A hand pump is provided in the system to act as an emergency device and to charge the accumulator on the ground. Because this hand pump must act as a safety device, it should not be connected into the power system but should go directly to the pressure manifold. Therefore, in order to charge the accumulator on the ground an accumulator-charging valve must be provided, which, when open, connects the hand-pump pressure line to the accumulator.

There are no units in the DC-3 system that utilize extremely close fitting valves that might be worn rapidly by dirt in the hydraulic system, and thus it was not necessary to provide an extremely fine filter. A filter was provided on the DC-3 that is capable of straining out all particles larger than 0.015 in. in diameter, because particles larger than this might possibly jam the pumps.

The engine section forward of the fire wall is designed to be quickly interchangeable. In view of this requirement, it was desired to incorporate valves at the fire wall that when disconnected would automatically shut off the engine pump lines to obviate the necessity for draining the hydraulic system whenever an engine was changed.

A hydraulic automatic pilot is used on the DC-3, and the fluid is stored in the same reservoir as that used for the hydraulic system. The hydraulic-system pump is mounted on one of the airplane engines, and the automatic-pilot pump is mounted on the other engine. In order that the automatic-pilot system could be operated in flight after the failure of its pump and the hydraulic system could operate for landing and take-off after failure of its pump, a pump change-over valve was incorporated in the system. This valve was so arranged that when it is in its normal position the flow from the left-engine pump is directed to the hydraulic system and the flow from the right-engine pump is directed to the automatic-pilot system. When the valve is thrown to its alternate

position, the flow is reversed. That is, the right engine is connected with the hydraulic system, and the left engine is connected to the automatic pilot.

In order to prevent pump failure in case of failure of the automatic pressure regulator a safety valve is incorporated in the system. This is simply a relief valve set at a pressure somewhat higher than the normal system pressure.

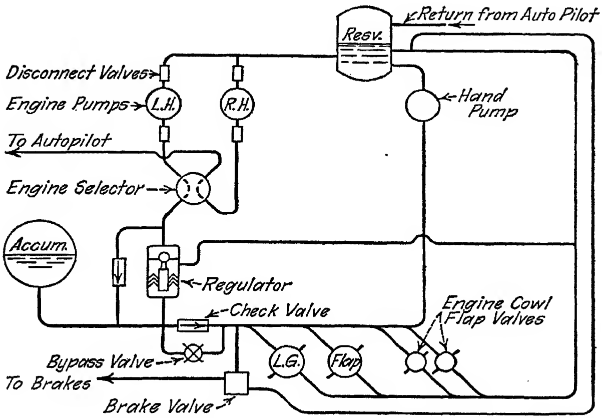


FIG. 60.—Schematic diagram of the power system of the DC-3 airplane.

Because in this airplane the four-way valves are normally kept in the closed position and a large accumulator is provided, it was not considered necessary to use a selective relief valve to ensure that fluid under pressure was always available to the brakes.

Because the automatic-pilot system turns approximately $\frac{1}{5}$ hp. into heat at all times, which keeps the hydraulic system at a temperature on the order of 50°F. above outside-air temperature, it was not considered necessary to provide any means of temperature control in the hydraulic system. There being no units in the hydraulic system that must operate at high altitudes, when the atmospheric pressure available to force oil from the reservoir into the pump is low, it was not considered necessary to provide a supercharged reservoir.

The schematic diagram of the power system down to the directional control valves can now be drawn. This is shown in Fig. 60.

It now becomes necessary to determine the line sizes and the sizes of relief and other valves in the system. The lines in the power system are ordinarily quite short, and therefore losses in these lines are determined more by fitting losses caused by turbulence than by viscous losses. To keep fitting losses down to a reasonable value it is only necessary to keep the velocity of the flow in the lines down to 10 ft./sec. or below. Any long lines in the power system should be designed with reference to pressure drop rather than velocity. Pump suction and pressure lines are ordinarily the only lines that fall in this category. The pressure

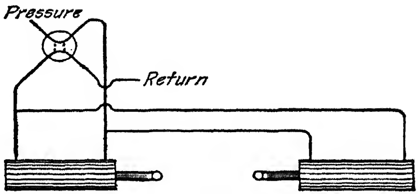


Fig. 61.—Schematic diagram of the landing-gear subsystem of the DC-3 airplane.

drop in the pump pressure lines should be kept down in the interests of reducing the load on the pumps to a value of not over about 5 per cent of the system pressure, which in this case would be about 40 lb./sq. in. The pressure loss in the suction lines must be sufficiently low so that under normal operating conditions the suction at the pump intake port does not exceed 10 in. of mercury. This requires relatively large suction lines in comparison with other lines in the airplane. Most of the pressure lines in the DC-3 are ½-in. tubing, but the suction line is ¾-in. tubing.

The necessary data are now available to put the line sizes on the schematic diagram of the power system, so that this is now complete. The next step is to determine the special requirements of each subsystem and draw their schematic diagrams, from the valve onward.

On the DC-3, special requirements for the landing-gear retracting system are few, for the airplane has no "up" latches, the "down" latches are mechanically operated, and there are no doors around the landing gear. The diagram of the landing-gear system is as shown in Fig. 61.

The line sizes for this system can be calculated, for the following figures are now known. The normal cylinder load required for

retraction is known from the load-stroke curve. The size of the cylinder has been previously determined, and the flow required also has been determined. The pressure drop available, or the loss in the lines, is the difference between pressure available and the average pressure required. The line size should be determined by picking a size that, at the lowest temperature at which the system is expected to operate at the required speeds, will not have a pressure drop exceeding that available.

As an example, the landing-gear retracting lines for the DC-3 airplane will be designed about as follows: On the assumption that

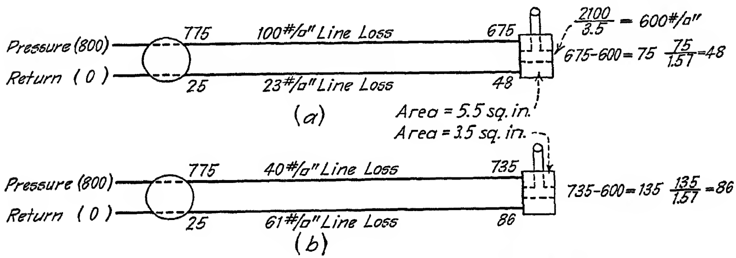


FIG. 62.—Line-loss diagram, DC-3 landing gear.

the load on the strut is about 2100 lb., the difference between the load, *i.e.*, pressure times area—on the retracting side, and the load—*i.e.*, back pressure times area—on the extending side of the pistons would be 2100 lb. Because the back pressure acts on a larger area, it cannot simply be said that the pressure on the retracting side minus the back pressure equals the effective pressure acting on the piston. Instead, effective back pressure must be used, which is equal to the actual back pressure times the ratio of areas, which equals in this case 5.5/3.5, or 1.57. A value for the pressure drop in the power system between the pressure-regulating device and the four-way valve will also have to be assumed. In this case, 25 lb. is a reasonable value. Likewise, a value for back pressure between the valve and the reservoir, which will also be taken as 25 lb., must be assumed. When the gear is being retracted, it is clear that the flow out the return line will be greater than the flow in the retracting line in a ratio of 5.5 to 3.5. The line loss will now be divided between the retracting line and the extending line by making a diagram similar to that of Fig. 62.

In Fig. 62 (*a*) is a first approximation to division of pressures

between the retracting side of the valve and the retracting side of the operating cylinder (the difference being the line loss) and the extending side of the valve and the extending side of the cylinder (the difference again being the line loss), which, as was previously pointed out, must be greater than the line loss in the retracting line. This first approximation obviously has too much line loss in the retracting line. In (*b*), the second approximation, the line-loss difference appears approximately correct, and line sizes can now be calculated from these line losses, the length, flow, and temperature being known, by using the line-loss chart (Fig. 16). In using this chart, it is necessary to assume a line size and calculate the line loss. If the line loss comes out too low, a smaller line size should be selected and the pressure drop again determined from the chart. If the pressure drop comes out too high, a larger line size must be chosen for the second attempt. This calculation need not be extremely accurate, for line sizes vary by wide steps and the nearest larger line size must be chosen. These same calculations should be run through for all long lines in the system and for operation of all the subsystems in both directions.

After all line sizes have been determined, it becomes necessary to set the relief-valve pressures. The relief-valve pressures should be as low as possible to prevent high pressures and high stresses in the system. However, they must be set sufficiently high so that they will not allow movement of the operating system under normal conditions. That is, they must not allow the landing-gear system partly to extend under any load factor less than the applied load factor for that airplane, and similarly for all other subsystems. It is permissible, of course, for the relief valve to relieve abnormal conditions such as load factors above the applied, and pressures built up by temperature expansion. It is not practical to set a relief valve closer than about 200 lb. or 10 per cent above normal operating, or system, pressures. This is necessary to prevent leakage in case of maladjustment of the regulating valve or the relief valve.

In cases where the landing gear is mechanically latched in the retracted position, the relief valve setting is usually determined by the minimum practical pressure differential above system pressure.

The design loads and pressures throughout the hydraulic system can now be determined. The maximum loads are determined by accelerated flight, maximum airload, and other maximum load

conditions. When operating cylinders are at their end positions, the loads tend to force them against their stops or latches. These loads cannot build up pressures in the hydraulic system but merely act as structural-design conditions for the operating cylinder and its attachments. Loads along the travel of the cylinders, however, do build up pressures in the operating system. Maximum loads can occur under either or both of these conditions. Relief pressures affect the lines, operating cylinders, valves, and all other parts of the system that are relieved by each relief valve.

Each of these three loads, operating loads, relief loads, and maximum loads, must be multiplied by the proper factor and the highest of the resulting loads taken as the design load for the particular system under consideration. These factors are determined from the outline below and are chosen to allow a sufficiently high margin in operating conditions to take care of fatigue strength of the various parts of the system. When relief pressures and maximum pressures are reached only rarely, only the yield strength of the materials need be considered.

Factors of Safety

All parts should be analyzed for the highest of the design loads below, ultimate strength of material being used.

1. *Frequently encountered working load* (multiply actual load by 2.5 to obtain *design load*). A working or operating load encountered during normal operation, possibly 25 operations or more per day or continuous loads such as
   - *a*. System pressure loads.
   - *b*. Bungee loads.
   - *c*. 1*g* landing or flight loads.
   - *d*. All flight-control applied loads.

2. *Rarely encountered working load* (multiply actual load by 2.0 to obtain *design load*). A working or operating load encountered during normal operation but relatively few times per day, possibly less than 25.
   - *a*. Air loads on doors, etc.
   - *b*. Relief pressure loads.
   - *c*. Retracting mechanisms and controls.
   - *d*. Emergency units.

3. *Stress-analysis design load* (obtain from stress analysis). An arbitrary design load or a structural design load such as
   - *a*. Arbitrary load (handle load, towing load, etc.).
   - *b*. Accelerated flight or landing loads.

4. Strength of lines. Lines subject to accelerated flight loads shall not be stressed above the yield point under limit (applied) load conditions regardless of relief-valve setting. This need not be checked unless accelerated flight pressure exceeds 2½ times system pressure.

   Lines shall be capable of withstanding 2½ times the operating or 2 times the relief pressure at the yield strength of the material and 5 times the operating or 4 times the relief pressure at the ultimate strength of the material. This applies also to flexible hose.

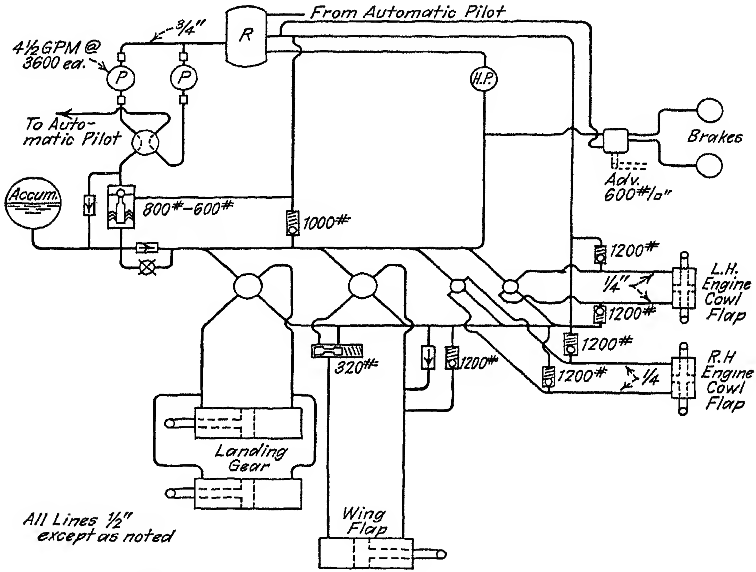


FIG. 63.—Schematic diagram of the hydraulic system on a typical transport airplane.

The factors for use in lines are higher than those required for operating cylinders because allowances must be made for flattening of the lines at bends, reductions in strength at flared ends, etc.

All these factors are sufficiently high so that the ultimate strength of the material can be used in the calculations. The actual design of the sizes of the various members of the system—*i.e.*, the cylinder-wall thickness, thickness of end caps on cylinders, etc.—will be described in Part III, Unit Design. The line size, however, can be determined now, by taking the highest design pressure, including the proper factor, and calculating the stress in the line by the

formula $S = Pd/2t$. In this formula, S is the stress, P is the pressure, d is the inside diameter, and t is the wall thickness. When calculated by this formula, using the design pressure, the stress must not exceed the ultimate strength of the material.

The necessary information is now available to draw a complete schematic diagram of the system showing the units and their hookup in both the power system and all the subsystems and all pressures and line sizes. The sizes of the operating cylinders and of the lines that connect to the valves, regulating devices, etc., which determine the sizes of these units, are also known. A complete schematic diagram for a typical transport airplane is shown in Fig. 63.

Now that the system has been determined, it is necessary to proceed to the detail design of the units making up the system, which will be discussed in Part III.

PART III

UNIT DESIGN

CHAPTER VII

DESIGN OF BASIC UNITS

Unit design will be covered in three chapters, the first of which will take up the design of the units comprising the basic system, including reservoirs, pumps, valves, operating cylinders, and lines and fittings.

Before designing any hydraulic unit, certain factors must be known that ordinarily are calculated at the time that the system is designed. (1) The type of unit to be used must be known. This is usually described as "similar to" some previously designed unit. (2) The size of the unit must be known, including its displacement, if it is a pump or operating cylinder, and the size of the passages or size of connecting lines if it is a valve or control unit. (3) The controlling pressures must be known, including the normal operating pressure and the design pressure. (4) It is necessary to have some idea of the surrounding structure in order to provide the proper attachments and to locate ports in such locations that they will not interfere. (5) All special requirements of the unit, such as the inclusion of latches, must be known. There are certain special requirements that apply to each unit. These will be given as the design of each unit is taken up.

Any standard parts available must also be known. These include such items as bearings, packings, check valves, and fittings, which are usually procured from outside sources or which have been standardized by such agencies as the SAE or the military services. The use of standard parts simplifies design considerably in that it obviates the necessity of designing special packings or other special parts for each new unit design. It also simplifies maintenance in that the parts which might need to be replaced when the unit is overhauled, including packings, bearings, etc., are standard parts and thus are readily available.

Standard packings are available in two forms, the V packing ring shown in Fig. 64 and the cup packing shown in Fig. 65. A third type, which is a simple round ring of synthetic rubber, is also coming into general use.

The first two have both been adopted from other industries, the V type from various steam cylinders, water pumps, and similar devices and the cup type of packing from the packing used in automotive brakes. The V packing is commonly used where long

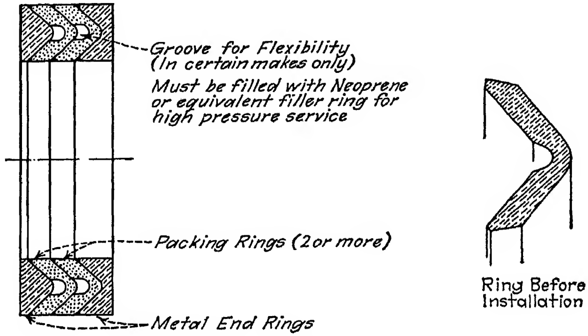


FIG. 64.—V-type packing.

wear and long life are desired and where the clearance of the back-up ring cannot be held to extremely close limits. The disadvantage of this type of packing is that it requires more space than the cup packing and is therefore not so well suited for installation in small units such as valves. This packing is used almost entirely in operating cylinders.

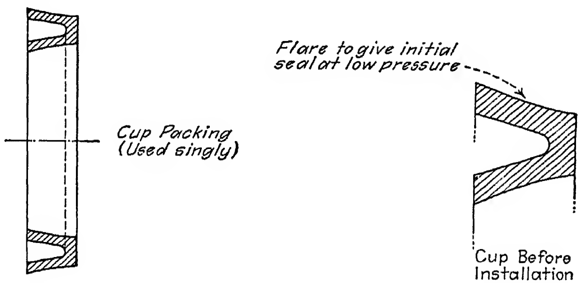


FIG. 65.—Cup or U packing.

The cup packing requires less space for installation but requires closer tolerances around the back-up ring to ensure satisfactory life and must be mounted in relatively rigid parts because expansion of the surrounding parts when under pressure will otherwise

increase the clearance around the back-up ring to a point where the life of the parts will be unsatisfactory.

Round-ring packings are used both in cylinders and in valves. They present in general the same advantages and disadvantages as cup packings, with the further advantage that one ring will seal in both directions.

Other standard parts such as ball bearings and bolts can be obtained from various manufacturers' catalogues and the Air Corps Standard Parts Book.

Of the various units that make up the hydraulic system, one which is common to them all is the reservoir. The type of reservoir that is used on almost all modern aircraft is constructed of welded aluminum-alloy sheet. It may be of various shapes, depending on its location in the airplane. The shape that is best from a hydraulic standpoint is the vertical cylindrical tank because when a high flow is circulating through the tank the oil can be introduced into the tank tangentially, thus giving a smooth rotating flow in the tank with a minimum of turbulence and therefore a minimum of foaming.

Size is determined by the requirements of the airplane. The rule for determining the capacity of a hydraulic reservoir is as follows:

> Capacity. A fluid reservoir shall be provided, having a capacity of at least 150 per cent of that sufficient to provide for changes in volume due to temperature change of 100° F. and to changes in internal volume of all operating mechanisms including pressure accumulation. The total amount of fluid thus required shall be provided above a point at least 2 in. above the top of the reservoir outlet which leads to the power-driven pump or pumps. A sufficient volume of fluid shall be provided below this outlet, but above the reservoir outlet which leads to the auxiliary hand pump, to permit complete actuation of all mechanisms essential to making a safe landing (except the landing gear when it can be extended in emergency without the use of hydraulic power, and provided that fluid does not flow into the landing-gear cylinders during such emergency extensions), assuming that no fluid is being returned to the reservoir during such actuation.

Reservoirs are normally tested to a pressure of about 5 lb./sq. in. to check them for leaks. This pressure need not be considered in the design of the reservoir unless the reservoir has relatively large flat areas. A circular reservoir with domed ends and in the sizes

normally used is capable of taking up to 30 or 40 lb. pressure per square inch without yielding. When the reservoir is supercharged to provide fluid at the engine pumps at high altitudes, the reservoir must be designed for a pressure sufficiently above the supercharging pressure to provide a reasonable margin of safety.

The structure surrounding the reservoir determines the position of the ports on the reservoir except that the vertical position of certain ports such as the engine-pump outlet and the hand-pump

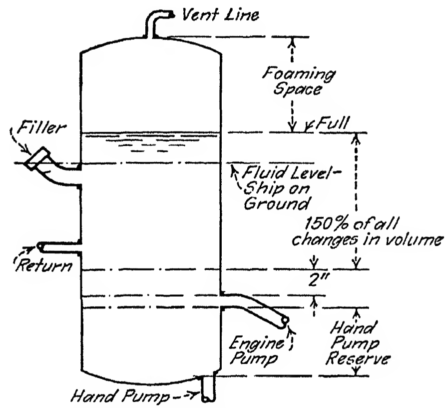


FIG. 66.—Typical reservoir showing fluid levels.

outlet must be determined from the position of the fluid levels. These fluid levels are shown on the typical reservoir shown in Fig. 66. The location of the levels is determined by the requirements of the preceding paragraph on reservoir capacity.

The supporting structure for the reservoir normally consists of a padded support, on which the bottom of the reservoir rests, and a felt-lined clamp or strap to brace the top of the reservoir.

Special requirements for reservoirs include proper port locations, the incorporation of a filter in some cases, and some sort of level-measuring device such as a gauge glass or a dip stick.

The reservoir is nearly always constructed of welded aluminum and is similar in construction to aircraft fuel tanks. The ports usually consist of aluminum castings welded to the reservoir and tapped for pipe threads or, in more recent designs, for the straight threads used with the swivel type of fittings.

There are two general types of hand pump used in aircraft, the double-ended type shown in Fig. 67 and the single-suction type shown in Fig. 68.

In the double-ended type, each end of the pump is a separate single-acting pump. In the single-suction type, when the piston travels to the right oil is sucked into the left end of the pump and fluid is forced out the outlet port. When the piston is moved to the left, oil in the large left-hand chamber is forced

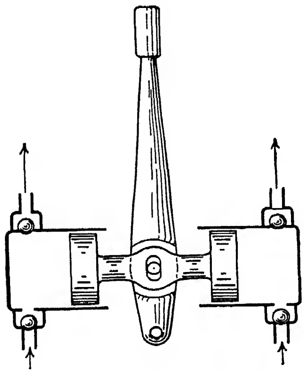


Fig. 67.—Double-ended hand pump.

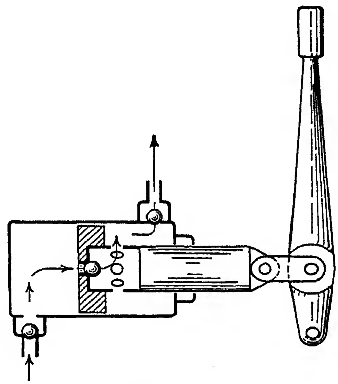


Fig. 68.—Single-suction type hand pump.

through the check valve in the center of the piston into the smaller annular chamber around the piston, the excess going out the pressure port. Only two check valves are necessary in this type of pump, an intake check valve and a check valve in the center of the piston. Because the hand pump is an emergency device, it is also desirable to incorporate a check valve at the outlet port, for otherwise after failure of the intake check valve the pump would not pump on either stroke, whereas, if an outlet check valve is incorporated, any one check valve can fail and the pump will still pump on at least one stroke. This not only is safer but also gives warning of approaching complete failure.

The displacement or size of the pump is determined by the maximum pressure that it must deliver under normal operating conditions and by the allowable handle load. For instance, in a pump that must deliver 1000 lb. of pressure with a 50-lb. handle load and

a handle travel of 16 in., the work available is 50 × 16, or 800 in.-lb. Since the efficiency of the pump will probably not exceed 90 per cent, only 720 in.-lb. of work can be done on each stroke if the handle load is not to exceed 50 lb., or a total of 0.72 cu. in. per stroke at 1000 lb./sq. in. 1000 lb. × 0.72 cu. in. = 720 in.-lb. of work. Since in a single-suction type of pump the intake stroke must pump sufficient fluid for both outlet strokes, its displacement in this case would be 1.44 cu. in., and the displacement of the annular chamber around the piston rod would be 0.72 cu. in.

A pump in which the ratio of stroke to bore is close to 1 to 1 is likely to be lighter in weight and more conventional in design than a pump with a very long stroke and a small bore or an extremely short-stroke large-bore pump. One pump in common use, which has approximately the displacement just given, has a 1-in.-diameter piston rod, a 1⅜-in. bore, and a 1-in. stroke.

The maximum pressures used for the structural design of the pump, including wall thicknesses of the cylinders and thicknesses of the cylinder heads, depend on the system to which the pump is connected. If there is no relief valve in this system, the pump must be designed for the highest handle load that can be applied to it. This is on the order of 250 lb. and would produce a pressure in the pump cited above of about 5000 lb./sq. in. However, most systems incorporate relief valves. If the relief valve in the outlet line is set to 2000 lb./sq. in. and the ordinary factor of safety of 2 is used, then the design pressure would be 4000 lb./sq. in. and the pump would be designed for this pressure. The stress in the cylinder walls, which should not exceed the ultimate strength of the material, will be determined by the usual formula $S = Pd/2t$. The stress in the flat cylinder head can be determined by calculation but can more easily be determined by referring to charts in which D/t is plotted against pressure for materials of various strengths. Such a chart is given in Fig. 69.

The surrounding structure determines only secondary features of the pump design. It determines the type and location of the pump attachments, handle arrangement, and port locations. The two types of pump attachment now in common use are through bolts and tension bolts. The shear-bolt attachment generally results in a satisfactory design, but it may loosen after repeated pumping unless the bolts are kept tightened. In the tension-bolt arrangement, looseness in the boltholes cannot cause wear, for

loads are taken in tension in the bolt, which can be tightened to eliminate all lost motion. The pump handle should be so located

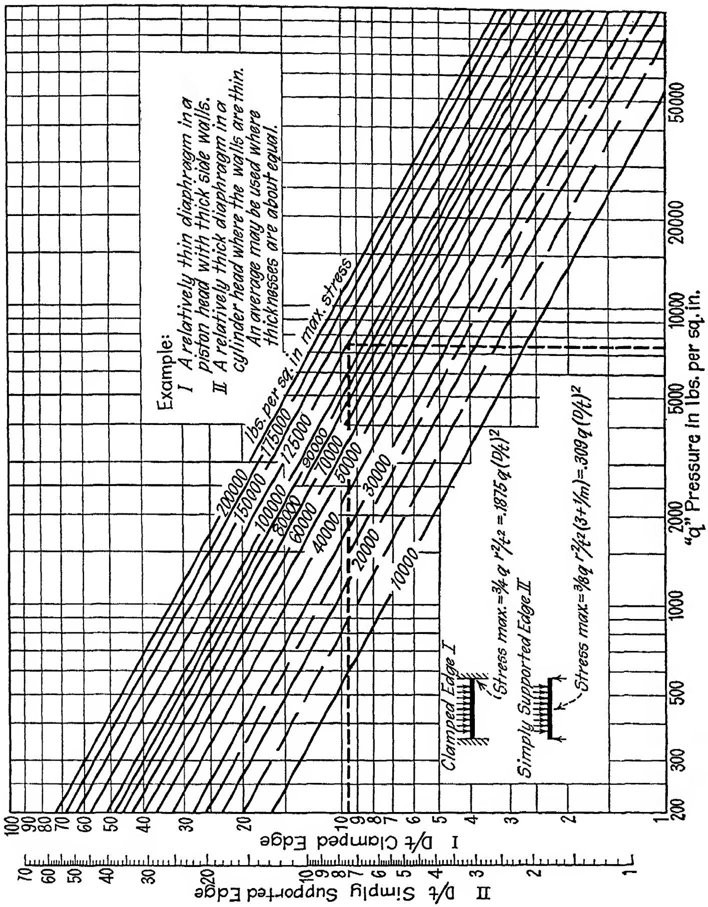


FIG. 69.—Stress in uniformly loaded circular plates, such as cylinder heads.

in the airplane that the direction of motion of the grip is approximately in line with the forearm of the person pumping. That is, the handle axis is approximately at right angles to the forearm of the person pumping. Any other arrangement makes it more

difficult to pump, and the handle loads must be correspondingly reduced if the pump is to be operated without excessive exertion.

Port locations must be determined more from hydraulic considerations than from the surrounding structure, for outlet ports must be on the top of hand pumps to permit any trapped air to be pumped out. Also, the location of ports along the travel of the piston is determined by the design of the pump. One special requirement of hand pumps is that, where there is a relief valve in the outlet line for the purpose of backing up pressure to ensure brake operation or for some similar purpose, the hand pump should be capable of pumping air to a pressure above the relief-valve setting. If this is not done and air gets into the pump, a condition may be encountered in which the pump will simply reciprocate, compressing and expanding the air, but not freeing itself of air, because the air cannot be compressed to a sufficiently high pressure to go past the relief valve.

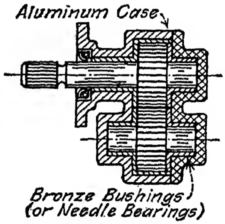


FIG. 70.—Typical gear pump.

Pumps are made from various materials. Usually the bore is steel to provide maximum wear resistance. This may be accomplished by providing a steel body or by using a cast or forged dural body with a pressed-in steel sleeve. The piston rod is almost invariably steel with a hard chrome-plated surface to minimize wear resulting from dirt, which might adhere to the oily piston rod. The handle socket is usually cast or forged dural, and the handle itself is usually a piece of dural tubing.

There are two types of engine pump in common use, gear pumps and piston pumps. Gear pumps as shown in Fig. 70 are familiar to most engineers, and no description of them should be necessary beyond saying that oil travels from the intake port around the outside of the gears to the outlet port and cannot return through the meshing teeth in the center of the pump. These pumps are not suitable for pressures above 1000 lb./sq. in. unless special devices are incorporated to reduce the leakage which at high pressure occurs past the ends of the teeth and past the sides of the gears.

Piston pumps such as the one shown in Fig. 71 can be much more closely fitted and are suitable for pumping much higher pressures, up to 3000 lb./sq. in. Operation of this type of pump is as follows:

When this pump is rotated clockwise (looking at the mounting flange) the pistons travel in and out with respect to the rotating cylinder block. When the pistons are traveling in, they force oil out the outlet port in the stationary valve block, and when they are traveling out, on the other side of the pump, they draw oil into the cylinders through the intake port.

The size of the engine pump is determined by the maximum flow requirement in the system and is usually expressed in g.p.m. at some specified speed. It may also be expressed as cubic inches

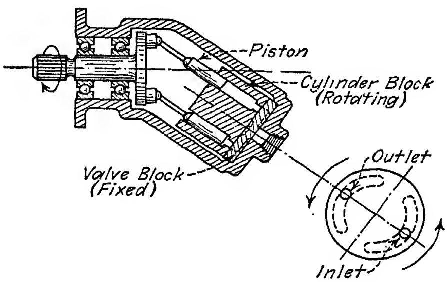


FIG. 71.—Typical piston pump.

per revolution. Pump speeds are usually 1 to 1½ times the engine crankshaft speed. There are no design pressure requirements for engine pumps; they simply must operate without excessive wear at their normal operating pressures.

The surrounding structure consists of the engine and its accessories. This determines the mounting pad and type of drive shaft of the pump, and surrounding engine accessories may determine the position or direction of the intake and outlet ports. The size of the ports is determined by flow considerations. As in most units, the flow should be kept below 15 ft./sec., and preferably lower in the suction port.

One special requirement that applies to engine pumps is the provision of a weakened section in the shaft. This will shear off and protect the engine in case of internal binding or seizing in the pump.

Pumps of the piston type are made into variable-displacement pumps by arranging the cylinder block and valve plate so that they can be swung into line with the shaft, the displacement being

thus reduced to zero. One type of pressure-regulated variable-displacement pump is shown in Fig. 72.

The use of this type of pump eliminates the necessity for any other pump-regulating means in the system; for when the pressure rises after the travel of some operating unit has been completed, the rising pressure forces the pump back to zero displacement, thus stopping the flow in the lines. Variable-displacement pumps must always pump their own leakage. Leakage in the pump, past the valve plate or cylinders, tends to reduce the pressure in

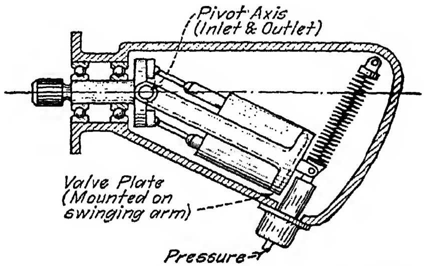


FIG. 72.—Typical pressure-regulated variable-displacement pump.

the outlet line, so that the pump then has to move to a partial displacement position sufficient to pump an amount of fluid equal to the leakage. This results in heating, for this fluid does no useful work. However, if there is not too much leakage through the pump, the heating does not exceed that generated when a constant-displacement pump operates at the back pressure necessary to force the fluid through the pressure lines and back to the reservoir.

Most aircraft pumps operate under pressure only a small percentage of the total flight time, for most aircraft incorporate a system of pump regulation using a constant-displacement pump and automatic by-pass valve. When the pump must operate under pressure continuously, the bearings must be larger and the pump must be designed more conservatively in order to give a satisfactory life.

Most aircraft manufacturers do not design or manufacture their own engine-driven pumps but buy them from manufacturers specializing in such units. Such data as size, pressures, engine attachments, and other special requirements must be provided the pump manufacturer if he is to design a satisfactory pump. These

are usually provided by writing a pump specification, which is sent to all the interested pump manufacturers, if a satisfactory pump cannot be chosen from manufacturers' catalogues.

Operating cylinders have been standardized in design during the past few years. Thus, most cylinders closely approach the design shown in Fig. 73.

The areas of the operating cylinder are determined by the loads that the cylinder must operate against and by the pressure available. The stroke is generally determined by the design of the mechanism that the cylinder must operate. When the strut must operate chiefly in tension, the piston rod should be made as small as possible; however, if it is made too small, the cylinder will not be a satisfactory piece of machinery. As a general rule, the piston-rod diameter should not be less than one-fiftieth the extended length of the strut.

The pressures, both operating and design, are normally determined when the system and its loads are calculated. The pressures determine cylinder-wall sizes, cylinder-head size, etc.; the external loads, which are applied to the cylinder at the end of its travel, affect the size of the end fittings.

The surrounding structure has considerable effect on the design of the strut. It determines the type of end attachments and has some effect on the port locations, which, however, should be as close to the top of the cylinder as possible, so that trapped air can be returned to the reservoir when the cylinder operates.

Special requirements for cylinders may include such features as bleed valves for removing air from cylinders, on cylinders that have to operate in synchronism with other cylinders where air might lead to erratic operation, or dashpots in the ends of cylinders for gradually slowing down heavy weights such as landing gears at the end of the stroke. Dashpots are not required to slow down the moving parts of the cylinder itself, for most cylinders move relatively slowly, at speeds usually not over 1 ft./sec. However, where a heavy landing gear, which may weigh 100 to 1000 times as much as the weight of the parts in the cylinder itself, must be stopped at the end of its travel, a dashpot may be required to bring this weight to rest gradually, in order to prevent damage to the inside of the cylinder or the attaching structure. A typical dashpot design is shown in Fig. 74. This dashpot operates by trapping a certain quantity of fluid inside the hollow piston head,

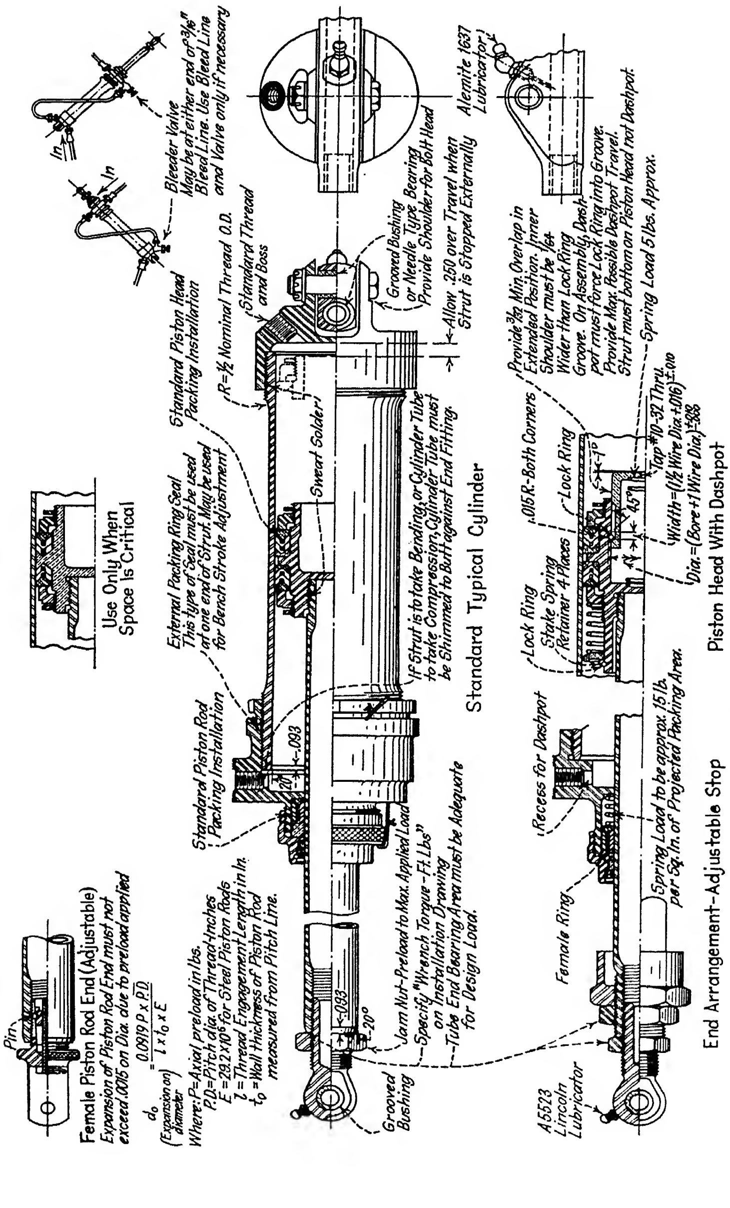


FIG. 73.—Typical operating cylinder.

which, at the end of the travel, must be forced out the small gap between the plug on the cylinder head and the piston head, thus offering considerable resistance to motion. Another type of dashpot is shown on the typical cylinder in Fig. 73.

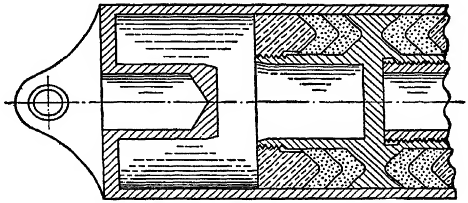


FIG. 74.—Simple dashpot.

The construction of most cylinders consists of a steel cylinder barrel, cast or forged aluminum-alloy ends, and a chrome-plated steel piston rod with an aluminum-alloy or bronze piston head. Packing is, in almost all cases, of the V type or the round-ring type.

Three types of directional control valves have been used in aircraft hydraulic systems, plug valves, slide valves, and poppet valves. Plug valves, such as that shown in Fig. 75, were used extensively some years ago but are not often used in modern hydraulic systems because of the high handle loads required and the leakage that rapidly develops as a result of wear between the metal parts.

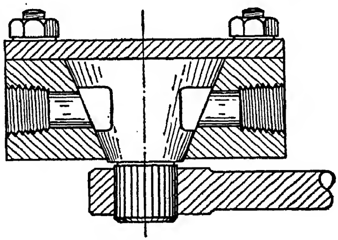


FIG. 75.—Typical plug-type directional control valve.

Slide valves, a typical example of which is shown in Fig. 76, may be used wherever leakage is not important but where extremely low handle loads are required. Such valves are usually designed to be in hydraulic balance; *i.e.*, the hydraulic pressures have no tendency to jam the piston or to move it in either direction. Because such valves depend on a viscous seal between the piston and the bore, they always have a certain amount of leakage, depending on the pressure and the viscosity of the fluid in the system. This leakage ranges from a few drops per minute in a

small accurately made valve using Sperry oil at moderate temperatures to 1 g.p.m. or more in large valves that are badly worn, using Sperry oil at high temperatures.

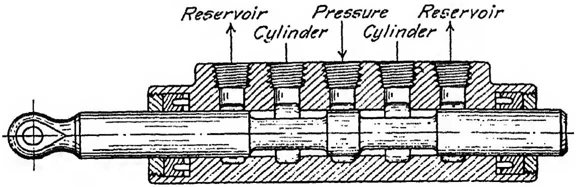


FIG. 76.—Typical slide-type directional control valve.

Poppet valves, a typical example of which is shown in Fig. 77, are the type of valve most commonly used in modern aircraft hydraulic systems. These valves will seal perfectly and will maintain their seal for a considerable period of time; *i.e.*, they do not wear rapidly. They are suitable for very high pressures and for thin fluids because the seal is created by metal-to-metal contact and the loads at the sealing surface are sufficiently high to produce a satisfactory seal, but not so high as to cause local failure. Such valves do not wear rapidly because the valve is raised off and lowered onto its seat and there is no sliding motion between the valve and the seat. The operating means may be a longitudinal cam or a rotating cam.

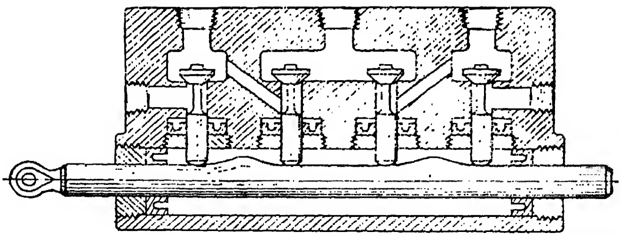


FIG. 77.—Typical poppet-type directional control valve.

The size of any one of these types of valve is determined by the size of the attaching line and the requirement that the pressure drop at the maximum rated flow should not be excessive. This usually results in a valve having flow velocities through its smallest passages—usually the passage between the valve and the seat—of not over 15 ft./sec.

Because of the fact that such valves are generally made from a block of material or, at any rate, have relatively heavy walls, the pressure strength is usually not a determining factor in the design of the valve. However, the handle load is directly determined by the load required to raise the poppet off its seat against normal operating pressure. Because most valves are designed as standard parts, the surrounding structure obviously can have no effect on the design of the valve. However, the selection of a standard valve or of a design, in the case of a special valve, may be determined by the surrounding structure to the extent that it may determine port locations and handle arrangement.

There are a few special requirements for valves. It is customary to provide a distinctive handle on landing-gear valves in order to reduce the possibility of inadvertent retraction of the landing gear through mistaking the landing-gear valve for the wing-flap or some other hydraulic valve. Some valves are also arranged so that the handle is returned to the neutral position by springs or by pressure rise. Such valves are not commonly used, however.

Small poppet valves are sometimes made by using a steel body with the seats machined as part of the body. The larger valves are usually made from a cast or forged dural body having inserted seats of a harder metal. The poppets are almost invariably made from hardened steel, although one recent design uses plastic poppets. Slide valves should have both the body and the valve piston made from hardened steel to prevent wear from particles of dirt in the oil. Also, if slide valves have both the body and the piston made of the same metal, clearances will not change with temperature changes.

The lines used to connect these various units may be of various materials, the commonest of which are 52SO and other soft aluminum alloys, stainless steel in either the annealed or quarter-hard condition, and 17ST aluminum alloy. 52SO lines have the advantage that the material can be easily bent and flared, which is necessary where the AC811, AN, or "Parker" type of flared line fitting is to be used. This material has the disadvantage that because of its low strength/weight ratio the wall thicknesses and weights become excessive when used in a high-pressure system. 52SO and similar alloys are not normally used when the operating pressure exceeds 1500 lb./sq. in.

Stainless steel in the annealed condition can be used for higher

pressures than 52SO, because of its higher strength, but it is relatively heavy because of its low strength when compared with its weight. Stainless steel in the quarter-hard condition is a suitable material for high-pressure lines. It is not suitable for low-pressure lines because the wall thickness required would be so thin that the tubing could not be satisfactorily bent or flared. However, when operating pressures approach 3000 lb./sq. in., quarter-hard stainless steel is lighter than 52SO by a considerable percentage. The ends of this tubing must be annealed in order to flare it successfully. This annealing must be kept inside the sleeve of the fitting, for otherwise the burst strength of the tube would be only the burst

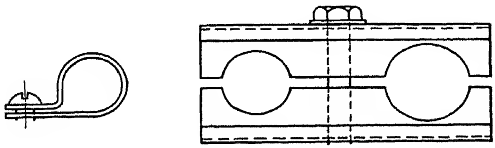


FIG. 78.—Typical line-support clips and blocks.

strength of the exposed annealed section. If the tube expands into the sleeve when abnormally high pressures are applied, no harm is done. In fact, the vibration strength of the line is increased, for then the vibration stresses are concentrated in a straight portion of the tube where it leaves the sleeve rather than at the base of the flare where the material has been distorted and thinned out by the flaring operation.

17ST has the serious disadvantage that, when used with the flared type of fitting, it must be worked in the annealed state and then heat-treated. 17ST can be bent in its hard state, however, even though it cannot be flared, and therefore is suitable when some type of flareless fitting is employed.

These lines are attached to the airplane by clips, typical examples of which are shown in Fig. 78. The loop type of clip is employed when a single line is to be attached to the structure. The clamping block is employed where a number of lines are to be fastened to the structure. These blocks may be made from bakelite or hard fiber, or they may be built up from a synthetic rubber or a Neoprene-cork compound backed up by aluminum channels. Both these types of clips must incorporate some sort of metal bonding strip because all lines in airplanes must be bonded to the structure

at short intervals to avoid radio interference. These clips are spaced along the lines at sufficiently close intervals to prevent the possibility of line failure through vibration. Long, unsupported lines may have a sufficiently low vibration period to cause them to be in resonance with some vibration of the airplane, such as that caused by the engine. When lines vibrate, they fail in fatigue at a relatively rapid rate. Recommended clamp spacings for clamps on hydraulic lines are given in Table II.

TABLE II.—RECOMMENDED SPACINGS FOR CLAMPS ON HYDRAULIC LINES

Size of line	Material	Maximum developed length, inches (measured along tube) between supports
3/16	Aluminum	12
3/16	Steel	14
1/4	Aluminum	13½
1/4	Steel	16
5/16	Aluminum	15
5/16	Steel	18
3/8	Aluminum	16½
3/8	Steel	20
1/2	Aluminum	19
1/2	Steel	23
5/8	Aluminum	22
5/8	Steel	25½
3/4	Aluminum	24
3/4	Steel	27½
1	Aluminum	26½
1	Steel	30

The fittings to which the lines are connected may be of several types. The type that is by far the most commonly used is the AC811, AN, or Parker type of fitting shown in Fig. 79. With this type of fitting the tube is flared and the seal is made by tightening the inside of the flare against the nose of the fitting with such force that the deformation that results makes a perfect metal-to-metal seal. In order to flare satisfactorily a metal must

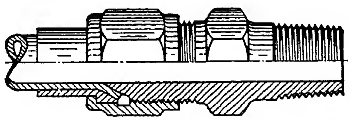


FIG. 79.—Typical flared-tube fitting.

have an elongation of about 30 per cent; this limits tubing materials to relatively low-strength materials.

There are several types of fitting on the market that do not require flaring of the tube, none of which, however, is in very common use. One type, which uses a flexible gasket to seal the tube end, merely crimps the end of the tube to take the tension loads. This type is illustrated in Fig. 80.

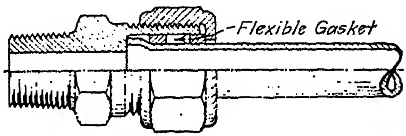


FIG. 80.—Flexible-gasket-type fitting.

This type of fitting, besides sealing against hydraulic pressure at very light wrench loads, is very satisfactory from the standpoint of vibration because vibration stresses are concentrated where the rubber meets the tube rather than at the base of the flare. Such fittings will normally last about twice as long as flared fittings when subjected to extreme vibration. Practically all fittings are designed to be as strong as the line to which they will be attached, so that pressure strength need not be considered in the choice of fittings.

The detail design of the units common to all hydraulic systems has now been discussed. The detail design of special units such as pressure regulators and brake control valves will be described in the following chapter.

CHAPTER VIII

DESIGN OF SPECIAL-PURPOSE UNITS

The design features of the common units in aircraft hydraulic systems have now been described. In this chapter, special-purpose units will be discussed, and examples of the procedure followed in their design will be given.

The need for special-purpose units arises whenever a new function is to be performed by the hydraulic system. As in the case of standard units, there are certain factors that must be known before the units can be designed. With special-purpose units, the first of these factors is the desired performance of the unit, or its operating requirements. The size of the unit must be given in terms of the flow of fluid through it and the loads that will be applied to it.

The pressures to which it will be subjected must be given, including the normal operating pressure and the maximum pressure. Sometimes it is necessary to take into consideration the surrounding structure or the structure to which the unit will be attached.

It is clear that any unit can be made up of only two components, as follows: valves, which control the flow of fluid; and pistons, which change fluid pressure into power, or vice versa. In general, piston design follows straightforward rules, but valve design depends largely on the design requirements. For instance, if leakage can be allowed through the valve and the forces available to operate the valve are low, then slide valves are indicated, whereas, if there is no leakage allowable and the forces available are fairly large, poppet valves must be used.

As an example of the procedure that is followed in the design of a special-purpose unit, consider first a brake control valve, one of the first of which was used on the Douglas DC-3 airplane. The requirement in this case was that a manual-brake system be duplicated as closely as possible. It was not possible to use a manual-brake system, because with an airplane of this weight the work required to operate the brakes made the force on the pedals prohibitively high with direct operation.

From the discussion of control systems, it is clear that there are three possible systems. The first of these is "load feel," the second is "position," and the third is a combination of the two. The problem of providing power brakes on the DC-3 was first approached by designing a system using both position and load feel. Figure 81 shows the design of a brake control unit providing both position and load feel. This unit operates as follows: When the pedal is depressed, the valve piston is pushed in, connecting the pressure line to the chamber behind the master cylinder and forcing it to the left; fluid is thus forced out the brake line, and the brake is applied. The pilot feels a load on the pedal equal to the brake

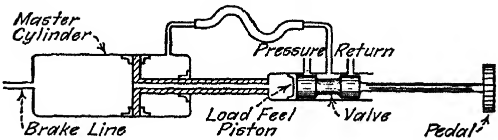


FIG. 81.—Brake-control unit incorporating position and load feel.

pressure times the area of the load-feel piston. The pedal position corresponds exactly to the position of the master-cylinder piston, except for valve travel which is negligible, and thus a manual-brake system is duplicated.

Because of the weight and cost of this system, it was decided to investigate the possibilities of using a simpler system that, though it would not duplicate manual operation quite so closely, still might be entirely satisfactory. The use of position control, in which a spring provided the pedal load, required a control unit that was the same as the control unit for the position and load-feel system of Fig. 81, except that the load feel was omitted.

Because this unit is obviously almost as complicated as the unit required by the position and load-feel system, a system using only load feel was tried next. This involved the use of only the valve portion of the control unit shown in Fig. 81. In order to get a movement of the pedals, it was necessary to incorporate a spring into the system to give a deflection of the pedals. The load, however, must be that produced by pressure rather than by a spring load. This system, the control unit for which is shown in its final form in Fig. 82, is the system which was finally chosen for use on the DC-3, for it seemed to possess many advantages over the other systems. Besides being simpler and lighter than either of the

other systems, it has an advantage over the position system in that it provides an indication of failure, for when the pressure in the brake lines drops, the load on the toe pedal drops correspondingly. It provides for simpler parking-brake arrangements, for all that is necessary in parking the airplane is to lock the toe pedals. After this the unit acts as a pressure-regulating device and keeps the pressure in the brake line constant regardless of expansion or contraction of the fluid in the lines, whereas, with the position type of

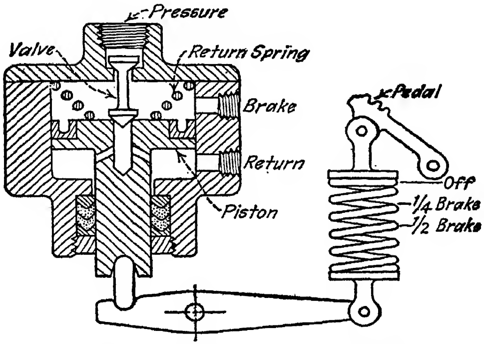


FIG. 82.—Brake-control valve incorporating load feel.

control unit, it is necessary to provide a spring-loaded expansion chamber somewhere in the system, for the fluid cannot get back through the valve but can only move the control unit.

The unit shown in Fig. 82 operates as follows: When the pedal is depressed to the half-pressure position, the piston moves up, carrying the valve with it and opening the pressure port. Fluid flows in through the pressure port and out to the brakes until the brake pressure rises to just above half pressure, when the piston moves down, compressing the spring slightly and shutting off the pressure port. The load on the pedal is now half the maximum pedal load, and the pressure is half the maximum brake pressure. When the brakes are to be released to the quarter brake position, the action is as follows:

1. The pedal is released to the new position.
2. This momentarily decreases the spring load.
3. The piston then moves down under the influence of the now unbalanced brake pressure, causing the return port in the piston

to open and allowing fluid to run out and brake pressure to drop until it reaches a value just below the spring load, when the spring extends slightly and closes the return port. The valve is now in equilibrium in the quarter-brake position.

Proportionality is ensured by the fact that the load felt on the pedal is due solely to the pressure on the top of the piston, which is, of course, the brake pressure. It should be noted that the spring does not affect the load on the toe pedal, only the movement of the toe pedal. If the spring is omitted, the toe-pedal travel is too short to feel natural and the kickback when the brake pressure comes up is noticeable.

Another example of a special-purpose unit is a surface-control booster. This unit was used for the first time on a recent experimental airplane. The requirement in this case was that a perfectly smooth load feel be provided and that position indication also be provided. In the original conception a requirement was also included that the unit be used as a control-surface lock by removing the load feel, *i.e.*, by making the unit an irreversible control and locking the unit positively in any position without allowing creeping from the locked position.

The requirement that load feel be perfectly smooth conflicted with the requirement that there be no leakage through the valve, in that slide valves, which give smooth operation, allow creeping, whereas poppet valves, which will seal perfectly, operate jerkily, for a high load is required to open them, which is reduced to zero as soon as the valve opens. The design requirements were changed in the process of design to remove the irreversible-control feature and substitute a method of control-surface locking that consisted merely of providing shutoff valves in the lines between the cylinder and the valve so that the oil in the cylinder would be positively locked there.

There is only one type of control system that will satisfy these requirements. That is the type which incorporates both position and load feel. There are still, however, several possible approaches to the problem. First, hydraulic load feel could be provided through the medium of an operating cylinder connected by lines to the main cylinder. This has the disadvantage of being very inflexible in design, for the cylinder area must be varied to change the load feel. This system also has the disadvantage that if air gets into the cylinder or interconnecting lines a loose coupling

is introduced in the system, which will cause the system to "motor," *i.e.*, to hunt, at a frequency and amplitude depending on the degree of such coupling.

Mechanical load feel has the advantage that the degree of load feel can be readily changed by changing leverage ratios in the system. Also, its rigidity is not likely to vary, for it depends upon deflection of metal parts; therefore, after sufficient damping has been incorporated in the system to eliminate motoring, there is

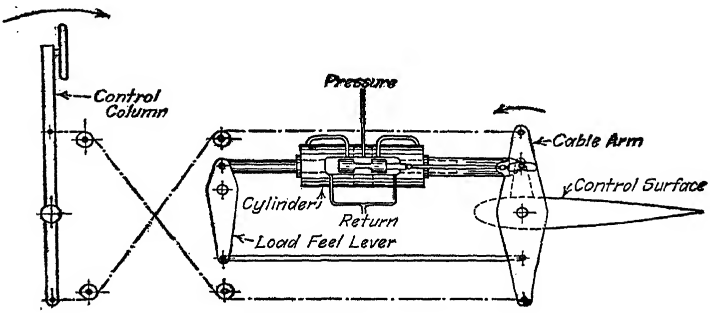


Fig. 83.—Surface-control booster.

little danger of motoring subsequently appearing owing to changes in coupling.

The position follow-up could also be either hydraulic or mechanical. However, in this case, only mechanical follow-up was considered for there was also the requirement that the pilot must be able to fly the airplane, although with very high control forces, after failure of the hydraulic-booster system. A mechanical follow-up system has the further advantage that it can never get out of synchronization.

The final design including the mechanical load feel is shown in Fig. 83. The operation of this system is as follows: When the pilot pulls the control column back, the cable arm, which is pivoted on the same axis as the movable surface, rotates to the left, pushing the valve piston and directing oil into the left end of the booster cylinder, thus forcing the cylinder to the left and moving the surface up. This action continues until the valve body, which is mounted on the cylinder, catches up with the valve piston and shuts off the flow of fluid. Thus the movable surface always follows the movement of the cable arm. Part of the load on the sur-

face, which equals the load in the cylinder, will be fed back to the cable arm where it can be felt by the pilot. The percentage of load that will be felt by the pilot is directly proportional to the leverage ratio of the lever connected to the cylinder end.

By following the principles by which these two problems were solved, a unit for any special purpose can be designed by any engineer familiar with the principles of aircraft hydraulics.

CHAPTER IX

HYDRAULIC-UNIT DESIGN PROCEDURE

The ultimate object of the unit design procedure covered in previous chapters is to produce a layout from which production, or shop, drawings can be made.

The procedure for making the design layout is usually divided into the following two major steps: (1) the sketch; (2) the layout. In designing any hydraulic unit, certain factors that govern design must be given before the sketch or layout can be started. These factors include (1) any special requirements; (2) some indication of the size of the unit, such as bore and stroke or size of passages; (3) pressures, both operating and design. In addition, the designer must know something of the surrounding structure.

Starting from these given factors and with a knowledge of similar designs or at least of designs incorporating some similar features, the designer can make a sketch, approximately to scale, of the unit. The areas, if a strut, or the passage sizes, if a valve, will have been previously arrived at. The diameter of a strut piston rod is usually made a minimum of one-fiftieth the extended length, to make the strut sufficiently stiff to prevent whipping. The sketch can readily be revised as the design proceeds, to make the component parts of the unit more easily machinable, easier to assemble, etc., *i.e.*, to refine the design until it is satisfactory.

When the sketch is satisfactory and the general design has been approved by the designer's supervisors, then the layout, which is a mechanical drawing accurately to scale, can be started. A rough idea of the required strength and sizes of the parts can be and should be obtained on the sketch. The member sizes are finally arrived at by running a stress analysis concurrently with the layout.

There are two types of loading that must be investigated during the design of a hydraulic unit; external loads on the unit, and internal loads caused by pressures within the unit. External loads include such loads as loads on operating cylinders from landing

loads, wing-flap loads, and handle loads on valves. Internal pressure generally determines the bursting strength of vessels subjected to internal pressure.

Perhaps the procedure of design for hydraulic units can best be explained by reviewing a typical example. In designing a landing-gear retracting strut for a typical bimotored airplane, the layout

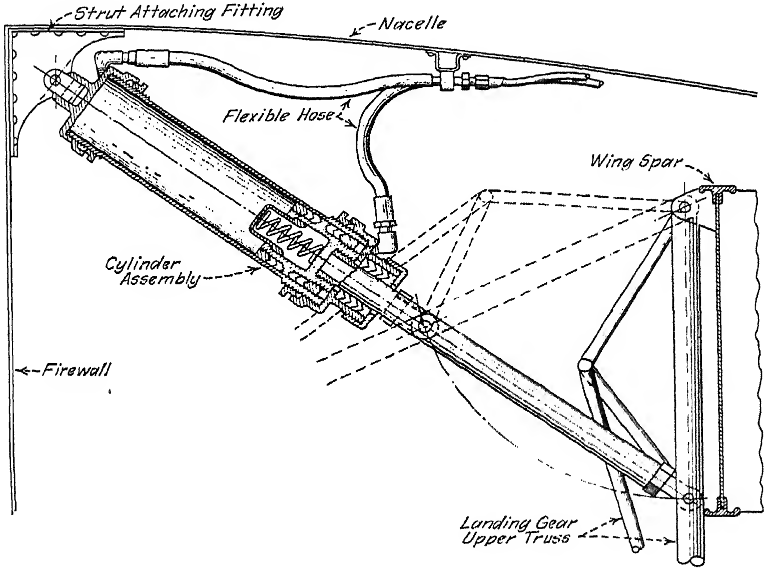


FIG. 84.—Layout—retracting strut.

for which would appear somewhat as shown in Fig. 84, the first step in making a sketch would be roughly to outline the surrounding structure, which in this case consists of the nacelle, with a strut-attaching fitting in its upper forward corner, and the landing-gear upper truss, to which the strut piston rod is attached. The strut would then be sketched in its proper place in the structure, by following in general the design shown in the typical strut in Fig. 73 (page 106).

Hoses for conveying the fluid to and from the hinged strut should also be shown on the sketch. After the sketch has been approved, the layout is started and proceeds along the same lines;

i.e., the surrounding structure is laid out and the retracting strut is then laid out on the structure drawing, the design shown in the sketch being followed.

The loads, both internal and external, are applied and the wall thicknesses of the parts determined. In this case, with an operating load of 2100 lb. tension, and a design tension load of 10,000 lb., for example, then the operating pressure will be 2100 divided by 3.6 sq. in. area (below the piston), which is 600 lb./sq. in., and the design pressure will be 10,000 lb. divided by 3.6 sq. in., or 2700 lb./sq. in. pressure. If the design compression load is 5000 lb., the corresponding pressure will be 5000 divided by 5 sq. in. area (above the piston), or 1000 lb./sq. in. Because the operating pressure even when multiplied by a factor of 2.5 is still below the design pressure of 2700 lb./sq. in., this design pressure will govern.

The following members must be investigated for the design tension load: the threaded joint where the end fitting screws into the piston rod; the threaded joint where the piston head screws onto the piston rod; and the threaded joints where the end caps screw onto the cylinder.

The following members must be investigated for the internal pressure caused by the design tension load: lower cylinder end, which is a flat head subjected to pressure and which can be analyzed from the chart, Fig. 69; also piston heads in some cases, though not in this particular example. The maximum internal pressure will also determine the wall thickness of the cylinder and may in some cases, if the piston rod is large in diameter and thin-walled, determine its wall thickness also.

In the case of the outer cylinder, the stress is calculated by the formula $S = PD/2t$. In the case of the piston rod, the stress is determined for external pressure from the chart, Fig. 85.

The compression load on the strut will determine the column strength of the piston rod. If the piston rod is taken as a column having the same length as the extended length of the strut between pin centers and having the section of the piston rod, an overconservative design results, for the strut is actually stiffened by the addition of the large cylinder over approximately half the length of the column. This can be taken care of with good accuracy by assuming a coefficient of fixity of 2 when analyzing the piston rod as a column, the piston-rod section being used for the full length of the extended strut.

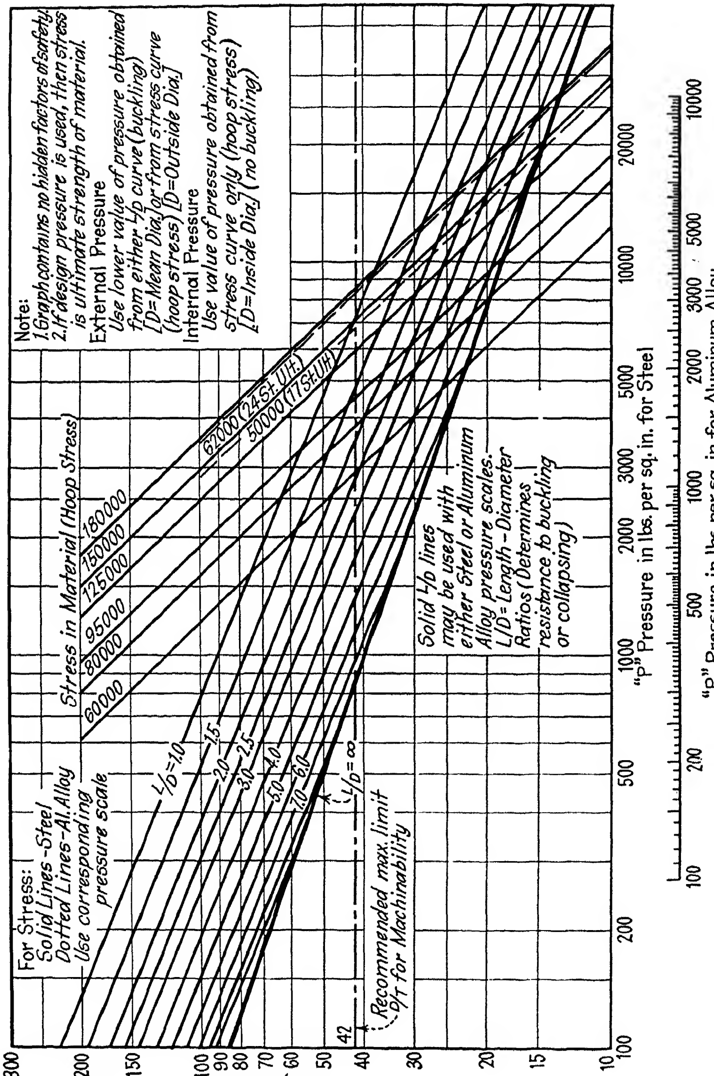


FIG. 85.—Critical pressures for tubes subjected to external pressure.

The end bearings will in this case be determined by tension, for that produces the maximum load. If the maximum tension load occurred with the strut fully extended instead of fully compressed or at part travel, it would, of course, produce no internal pressure in the strut and only the threaded joints would be designed by it.

Most aircraft companies have worked up from past experience certain sets of rules that must be followed in the design of hydraulic units. The rules that are followed by a typical aircraft company are in the form of a checking list for checking over layouts and are found in the company's Hydraulic Design Manual. These practices will be taken up in a subsequent chapter on Drafting on Aircraft Hydraulic Systems.

The stress analysis of aircraft hydraulic systems will not be discussed in great detail, for it can be worked out from the general principles given in a number of textbooks on general aircraft structure design and analysis, such as "Aircraft Structures," by Niles and Newell. Following the above procedure and design practices developed by experience should result in a satisfactory design for any hydraulic unit.

PART IV

DRAFTING, MANUFACTURE, INSTALLATION, TESTING, AND MAINTENANCE

CHAPTER X

DRAFTING OF AIRCRAFT HYDRAULIC SYSTEMS

Drafting of aircraft hydraulic systems starts with the design sketch, which was described in sufficient detail in the previous chapter. The next drawing is the layout, on which the design of the part is worked out.

Following the layout, detail or production drawings are made. These drawings are divided into three classes, as follows: detail drawings, which are drawings showing a single part; assembly drawings, which are drawings showing an assembly of several parts, but not usually including manufacturing details of each part; and installation drawings, showing the attachment of the parts or assemblies to the rest of the airplane.

Layout drawings are made from the sketch showing the design. They should also show the surrounding structure in the airplane, for they act as master drawings from which detail, assembly, and installation drawings will later be made.

The following points from the layout check list of a typical aircraft company give an outline of the drafting practices that should be followed in making layout drawings. This list covers only drafting practices that are more or less standard throughout the industry and does not include practices which have been discussed in general in the previous chapters or those which vary throughout the industry.

1. Clearances above, below, and on all sides of the part being laid out must be shown and noted, or reference made to the layout on which they appear, in such a way that the clearances can be readily checked. Maximum possible clearance should be provided to allow for future increase in size, additional lines, etc.

2. Both extreme positions and any critical intermediate positions of moving parts must be shown, and maximum possible clearance provided.

3. Fittings, hoses, and plumbing must be shown.

4. Locating dimensions, including vertical, side, and fore and aft dimensions from airplane reference lines, should be given.

5. All reference layouts and drawings must be listed by number and title.

6. Operating loads must be given on layout, including pressures, velocities, working or actual loads, allowable loads, available loads, maximum loads, and design loads for all mechanisms. Allowance should be made for friction, pressure drop in lines, etc. Note should be made of how the load was obtained to enable it to be checked.

7. Test data or reference to test reports or references to similar parts should be given for all designs of mechanisms, when available. Tests should be made on unconventional designs if none of the above is available.

8. All conditions of operation must be shown by schematic diagrams when necessary.

9. Materials and sizes should be noted unless obvious.

10. Heat-treatments, finishes, etc., should be noted unless conforming to general practice.

11. All necessary test and adjustment notes must be given on the layout, exactly as they should appear on the assembly drawing.

12. Strength calculations necessary for design of parts (including bearing pressures and approximate design loads) should be made directly on layouts.

13. Torque for preload should be given on the layout for the following:

Sweat-soldered joints.
Cup-seal joint.
Any special joint requiring preload.

14. Clearances, overlaps, etc., must be ample to allow the use of reasonable tolerances on the detail drawings. A clearance of 0.020 for each detail dimension affecting the clearance will allow ± 0.010 on half of the dimensions and $\pm 1/32$ on the remainder.

The last item on the list deserves more explanation. When detail drawings are made from a layout drawing, the detail drawing is "tolerance-checked" (as will be described more fully later on) to ensure that, even though all the tolerances act to reduce a clearance, overlap, etc., to the minimum, the parts will still go together. If small clearances are shown on layout drawings, then the tolerances of the detail parts, whose dimensions when added together determine those clearances, must be held to such close limits that the machining cost of the parts is increased.

Detail, or production, drawings are made from the layout drawing by making a separate drawing of each part and finally of the assembly, so that the parts can be made on various machines or in various parts of the shop without reference to other parts of

the same assembly. These drawings are made in accordance with drafting practices that are standard in all engineering departments. The detail drafting practices that are followed by a typical aircraft company in making production drawings for hydraulic parts are given in the detail check list from the company's Hydraulic Design Manual.

The more important of these points follow:

1. Detail drawings must agree exactly with the layout, which should be complete and in accordance with the layout check list. If it is necessary to change the layout, such change must be approved by the layout draftsman and checked against the layout check list. If time is not available actually to make the change, such change should be noted on the layout.
2. Layout numbers must be listed on all drawings.
3. Stock size listed should be one readily obtainable.
4. Material must have sufficient finish allowance for machining with raw stock tolerances at their extreme limits (tubing especially).
5. Notes should be on the drawing in the sequence in which the operations will be performed in the shop.
6. Concentricity notes and tolerances must be used on all diameters which must be concentric.
7. Groups of drilled holes must be dimensioned so that, with all tolerances adverse, parts can be assembled.
8. Special parts which resemble standard or purchased parts shall specify "Stamp part number here" or "Spec." if there is insufficient room for part number.
9. Surfaces that must be "square" or "flat and square for bearing" must be so noted.
10. Wrench torque for preload should be specified on the following joints:
    a. Sweat-soldered joints.
    b. Cup-seal joints.
    c. Any special joint requiring preload.
11. Assembly drawings provide a convenient means for checking the effect of the accumulated tolerances of the detail parts. The method of checking is as follows:
    a. Select the points where a small variation in the parts might cause interference or malfunction, bearing in mind that, where a clearance, overlap, etc., is controlled by six or eight dimensions, the parts will vary usually from $\frac{1}{16}$ (all dimensions ± 0.010) to $\frac{3}{16}$ (all dimensions $\pm\frac{1}{32}$); and calculate the clearance.

b. Points likely to cause trouble are thin sections, overlaps, clearances, etc., of ¼ or less, controlled by longitudinal dimensions.

c. Calculations should be kept in such form that they can be readily checked over, preferably directly on a black-and-white print.

In order to clear up more fully the procedure of tolerance checking, an example is given in Fig. 86.

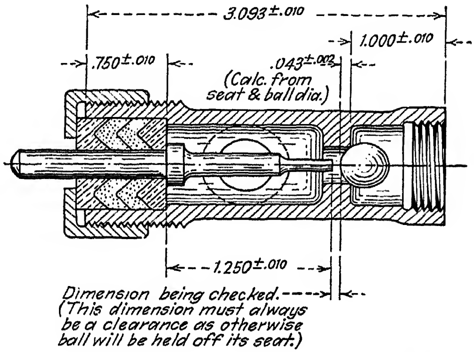


FIG. 86.—Example of tolerance checking.

Calculations for dimensions being checked are as follows:

Pin end to shoulder	−1.250	±0.010
Body shoulder to end	−0.750	±0.010
	−2.000	±0.020
Body end to end	+3.093	±0.010
	+1.093	±0.030
Body end to seat	−1.000	±0.010
	+0.093	±0.040
Seat to ball	−0.043	±0.002
Clearance	+0.050	±0.042

or from 0.008 to 0.092

Note that dimensioning the body from shoulder to seat directly would reduce 3 tolerances to 1 and would reduce accumulated tolerances from ±0.042 to ±0.022

Production drawings also make use of a large number of standard or coded parts, which makes it possible to design assemblies including packing without having to design a separate piece of packing for each unit. Parts commonly coded are replaceable parts such

as packings and bearings and parts that can be coded for size, such as the metal rings adjacent to packings, washers, and bolts.

The question often comes up in making detail drawings of the specification of surface finish. Some aircraft companies have adopted the practice of making a sample showing the finish that is desired on all pieces of high-class hydraulic machinery and furnishing all their subcontractors, as well as their own machining and inspection departments, with these samples.

The methods in most common use for actually measuring the surface finish are the various profilometers such as the Abbott (Physicists Research) and the Brush, which give the surface finish in terms of the root-mean-square (r.m.s.) average height of the irregularities of the surface in millionths of an inch, or microinches. For a good ground surface, this finish is between 6 and 10 microinches r.m.s. The National Aircraft Standards Committee recommends that finishes be specified from the following list: 2, 5, 10, 20, 50, 100, 500.

By following the principles outlined in this chapter and checking the finished drawing against the lists that have been given, it should be possible for an experienced draftsman to make satisfactory production drawings of aircraft hydraulic parts.

CHAPTER XI

MANUFACTURE OF AIRCRAFT HYDRAULIC UNITS

The manufacture of aircraft hydraulic units can be divided into four stages, planning, manufacturing, assembling, and testing. This chapter will cover the first three. Testing will be covered in a subsequent chapter.

Planning includes in general all the preparatory work that is necessary before actual manufacture of the parts is started. This includes scheduling, material ordering, and tooling.

The principal purpose of scheduling is to ensure that machines and man power are available to make the parts and assemble them in time to meet the required delivery date. This is also called the determination of "shop loading."

Material ordering must be done as early as possible in the planning of the job, because aircraft materials are becoming more and more difficult to obtain. Some of the more common materials can be obtained from supply houses' material stocks; but if large quantities or unusual materials are required, they must be obtained from the mill. Also, of course, forgings, castings, and other special parts must be obtained from the forging or casting plant.

As an example of the length of time required to obtain material, in normal times it takes approximately 10 to 12 weeks to obtain aluminum alloy in either bar or castings, both of which are used in aircraft hydraulic units. Alloy steels, such as chrome molybdenum, take 14 to 16 weeks to get, from the time the order is placed. Forgings in aluminum alloy take 12 to 16 weeks to obtain if a new die must be made and a few weeks less if a new die is not required. Steel forgings take 14 to 18 weeks when new dies are required. If the dies are very complicated or if development work may be required, these times may be increased as much as 50 per cent. Tube fittings, which are very often assembled into aircraft hydraulic units, require approximately 12 weeks for delivery after placement of order, even for standard fittings. Under wartime conditions, these times may be considerably extended.

The tooling required on aircraft hydraulic parts consists in general of the necessary fixtures to adapt standard machines, such as turret lathes, milling machines, etc., to the manufacture of the part to be made. Much larger quantities than are common in aircraft work are required before the same type of tooling can be made that is customary in automotive factories, where special machines are made to handle each operation in the manufacture of a part. As a rough guide, it may be considered that 10 complicated or 100 simple parts are required before any tooling is required, 1000 pieces justify fairly complete tooling (as aircraft tooling goes), and quantities on the order of 100,000 may be required before special machines can be justified.

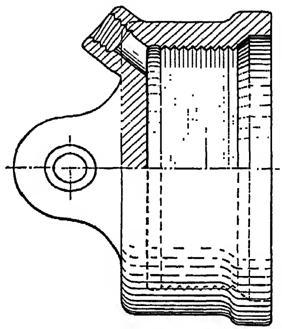


FIG. 87.—Typical cylinder head.

As an example of typical manufacturing operations, a few typical parts will be considered. Take first the cylinder head shown in Fig. 87. This might be made from a casting, either sand or permanent mold, a forging, or bar stock. Its machining would be dependent upon the machines available at the company that was doing the work. A typical routing for such a part would be as follows: (1) bore inside, face end on turret lathe; (2) mill lugs on milling machine; (3) drill bolthole and pipe connection on drill press; (4) mill threads on thread mill.

Cylinder barrels, a typical example of which is shown in Fig. 88, are usually made from steel tubing. Typical routing for such parts is as follows: (1) bore inside diameter, turn outside diameter on lathe; (2) thread ends on thread mill or thread grinder; (3) finish inside diameter on vertical hone.

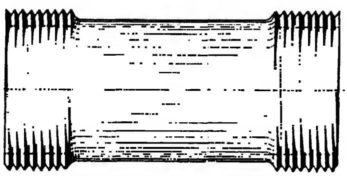


FIG. 88.—Typical cylinder barrel.

Piston rods, a typical example of which is shown in Fig. 89, are almost invariably made from steel tubing. These parts may be handled by one of two routings, as follows: (1) the grind, plate, buff routing; (2) the grind, plate, grind routing. For both these

the part is first centerless-ground, then threaded on either the thread grinder or the thread mill. For the grind, plate, buff routing, the part is then chrome-plated with a thin plating of one to 1½ thousandths thick per side, which is calculated just to bring the diameter of the part to the desired finished diameter, then buffed to smooth the surface. With the grind, plate, grind routing, after grinding and threading the part is plated with a heavy coat of chrome plate approximately 5 thousandths thick per side. The part is then reground, usually on a centerless grinder, to the desired finished diameter.

The disadvantage of the grind, plate, grind system is that, because a centerless grinder grinds the finish diameter concentric

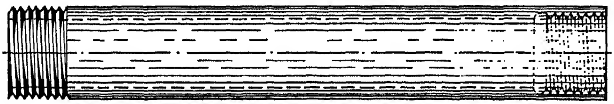


FIG. 89.—Typical piston rod.

with the outside, if the plating has been put on unevenly, after the finish grinding it may be much thinner on one side than the other. It also has the disadvantage that it is more expensive to apply this heavier coat of chrome plating, provided that both coats of plating are applied slowly, as they should be for maximum hardness.

It has the advantage, however, that the first grind job need not be so smooth as in the grind, plate, buff system because a new surface is put on by the final grinding, whereas, in the grind, plate, buff system, the initial grinding determines to a large extent the final surface finish.

Valve bodies, a typical example of which is shown in Fig. 90, can be made from bars or, for large parts, from castings or forgings. The usual routing for such parts is as follows: (1) Drill, bore, and tap on turret lathe. Here, reasonably good tooling is required, for there is usually a multiple series of holes, which must be machined. (2) Drill internal passages on the drill press. (3) Grind integral seats, if any.

Internal parts of valves and most other hydraulic units are generally made from bar stock on screw machines. Where large quantities are required, they are often made on automatic screw machines, where the bar is fed in one end of the machine and the

finished part comes out the other end, without any attention on the part of the operator.

The features of design that contribute to easy manufacture can best be understood by understanding the machines available to make the parts and the operations performed on them. As a general rule, it is more desirable to remove ample material, rather than take the chance of a rejected part because of insufficient cleanup in machining. It is often desirable to make parts from bar stock rather than castings even though two or three times as much material must be removed, because of the spoilage occasioned

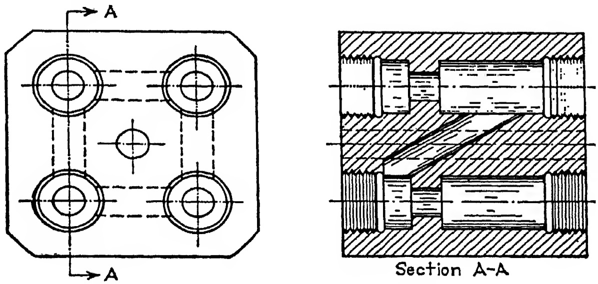


FIG. 90.—Typical valve body.

by leaky castings. On parts that are to be made on screw machines, on turret lathes, or in fact on most machines, it is desirable so to design the part that all the machining can be done from one end so that the part will be finished when it is cut off from the parent bar stock, to eliminate the necessity for turning the part around and chucking each part individually and concentrically in order to perform machining operations on the opposite end.

Assembly of hydraulic parts is simply a question of getting good mechanics who can work carefully on the relatively fragile parts in an aircraft hydraulic system—relatively fragile, that is, when compared with the parts of an automobile or a locomotive.

Most aircraft-parts manufacturing establishments seem to have come to the conclusion that it is economical to spend time on a very accurate job of manufacturing in order to save time and rejections on assembly. An example of this is the grinding of valve seats, which a number of companies have found desirable to reduce the time necessary to get a satisfactory seat. On a cone or ball poppet valve, both the diameter and the face of the seats are ground to

ensure a round hole and a sharp edge on which the valve can be seated very rapidly.

Good mechanics are required for installing pipe threads, tightening gaskets, and performing similar operations where over-tightening could cause excessive deflection with resultant binding and malfunction.

There are a few semimanufacturing operations that are normally performed by the assemblyman. These include, chiefly, staking, soldering, and lapping.

Staking is a locking operation that is performed by deforming one metal part into another to keep threads from loosening. It is customary practice in aircraft to stake the soft metal so that when the hard metal part is unscrewed the soft metal is formed back into its original shape. If a hard metal part is staked, it cuts, or deforms the soft metal parts when the parts are unscrewed, often making it impossible to reassemble them.

When parts that must be fluid-tight are to be soldered together, it is important that they be preloaded, while the solder is in a molten state, of course. This is necessary to avoid deflections, which may otherwise occur and which may crack the solder and cause leakage. If a part is preloaded sufficiently so that it will not deflect when operating loads are applied, then obviously the solder will not crack and the part will remain pressure-tight.

Where a metal-to-metal seal is necessary, parts must be lapped together. As an example, ball and cone valves are lapped onto their seats by the same technique that is used to lap valves in gas engines. There is, however, no common equivalent of the slide valves that are often used for controlling flow of fluid in aircraft hydraulic systems. In these valves, a piston is fitted to a bore with a clearance on the order of two or three 10 thousandths of an inch. This tolerance cannot be attained by conventional manufacturing methods, so that it is necessary to resort to hand finishing or lapping to get these clearances. The procedure that appears best for this purpose is to make the hole undersize and the piston oversize. The piston is then lapped down to approximately its desired finished size by reciprocating and rotating it at the same time in a female lap of a softer metal than the piston, which is coated inside with an abrasive. The hole is then lapped to give two to three 10 thousandths clearance with the piston. The male lap for the hole is generally made of softer metal than the part to be lapped and is

charged with abrasive by rolling it between two very hard flat plates coated with free abrasive. This makes the lap into a tool that is practically an inserted tooth broach. Such tools will enlarge a hole at the rate of about one 10 thousandth of an inch in 40 strokes. The combination of reciprocation and rotation ensures that the hole and the piston are straight and round.

After assembly the part is ready to be tested. Because testing applies to both assembly and installation, it will be discussed in the chapter following that on installation.

CHAPTER XII

INSTALLATION OF AIRCRAFT HYDRAULIC SYSTEMS

Installation consists of the installing of units, lines and fittings, flexible hose, etc., in the airplane. It will be considered in two parts, (1) design for installation and (2) the actual installation work.

In making a good installation of an aircraft hydraulic unit such as a cylinder or valve, it is necessary to consider the surrounding structure carefully, not only to ensure that no interference exists, but to allow sufficient room for mechanics to work in installing the part, bolting it in, tightening fittings, etc. In addition, certain requirements are necessitated by the fact that aircraft structure is not very rigid and sometimes not very exactly made.

It is desirable, unless the airplane is to be extremely well tooled, to put universal joints at the ends of operating cylinders so that misalignment at the end fittings will not cause binding in the hydraulic struts.

In the installation of valves, it should be remembered that a valve bracket that is amply strong to take the normal handle load is not necessarily rigid enough to give a good installation in the airplane. To ensure rigidity, it is necessary either to design for rigidity by tying in any free edges of sheet metal and avoiding the loading of sheet metal in bending or to put on very high arbitrary loads to ensure the parts being so strong that ample rigidity results. The first of these is to be preferred but requires a higher degree of engineering talent.

Installation drawings should specify operations in the order in which they will be performed in the shop, and also should specify preloads on important bolted joints, particularly such joints as end fittings on operating cylinders where lock nuts may loosen if not preloaded.

In designing line installations, it is common practice to allow the shop a certain amount of latitude in the routing of the lines, provided that they are approved by the engineering department.

Thus, such matters as bend radii and support spacing often become both shop and engineering functions. Lines should be supported at sufficiently frequent intervals so that they will not vibrate at a frequency that might be in resonance with the vibration frequency found in the part of the airplane in which they are installed, for otherwise early fatigue failure may occur. Support spacings should be in accordance with the spacings of Table II (page 111). These spacings are calculated to give a vibration frequency on the order of 100 cycles per second, so as to be above the major vibration frequency of the engine, which occurs at two times crankshaft speed, and which in the case of a 2400 r.p.m. engine would be 80 cycles per second. This is the highest frequency at which sufficient power is put into vibration to overcome the damping that results from the use of the usual Neoprene clamping blocks and the fact that the lines are filled with fluid, both of which tend to damp out the vibrations at the higher frequencies.

Bends should be made close to supports rather than midway between supports, in order to reduce the overhanging length of line, which has a tendency to vibrate. They should preferably be all of the same radius in any one line or, still better, should be the same radius for any given diameter of line throughout the airplane. This is desirable in order to avoid changing bending blocks on bending machines when the lines are manufactured. Lines that are to be bent by hand in the airplane, which includes usually only lines of about 1/4 in. diameter and smaller, may be bent on various radii or with sweeping curves if desired. Machine bending should be done on a type of bending machine that includes a mandrel inside the tubing to prevent flattening. Bending machines that do not use a mandrel flatten the tubing considerably, in fact so much that the fatigue strength of the line is reduced to about one-half the fatigue strength of a round line.

The minimum bend radius for tubing is usually taken as four times the outside diameter for tubes of 1/4 in. diameter and smaller and three times the outside diameter for tubes of 5/16 in. diameter and larger, measured from the radius center to the center line of the tubing.

With fittings of the AC811, AN, or "Parker" type it is necessary to flare the lines. This can be done by spinning or by impact. Impact flaring, in which a tool having the proper angle is driven into the end of the tube while the tubing is held in a properly

placed clamp block, is to be preferred over spinning because it work-hardens the tube less and usually results in a slightly thicker flare.

In installing fittings, a designer can prevent leaks by making parts incorporating male pipe threads from steel or brass, so that they can be tinned with soft solder, a ductile gasket being thus formed around the threads, which seals the fitting in spite of minor imperfections in thread form. Where the weight penalty of a steel fitting is prohibitive, the fitting can be made from dural and threaded with a female pipe thread, with a steel nipple tinned and screwed into it, the equivalent of a male thread fitting being thus formed.

In order to avoid pipe threads, fittings have appeared in aircraft hydraulic systems recently in which a hollow bolt is screwed into a fitting having an eye. This type of fitting can be located in any angular position. These fittings do away with the necessity for pipe threads, for one of the chief reasons for pipe threads is to enable the mechanic to locate an elbow fitting at the proper angle for installation. However, care must be used in tightening such fittings to ensure that gasket mating surfaces are flat and square and to avoid twisting off the hollow bolts.

Another type of fitting that eliminates the use of pipe threads consists of one on which the pipe thread has been replaced by a straight thread having a short unthreaded portion that comes opposite the top of the boss when the fitting is installed. The fitting is then made leakproof by screwing down a nut, previously installed on the fitting, to clamp a synthetic-rubber gasket, on the unthreaded portion of the fitting, between the nut and the top of the boss. The unthreaded portion is made the same diameter as the root of the thread to permit of screwing the gasket nut on from the lower end of the fitting. This fitting is easier to install and cheaper to manufacture and offers less restriction to flow than the "eye" type previously described.

When tube nuts are tightened on flared fittings, there is a tendency to thin the flare; in fact, it is necessary to thin the flare slightly, for the flared end of the tube acts as a gasket between the fitting and the sleeve or nut, to seal leakage. Wrench torques as specified in the Parker Service Manual should not be exceeded; otherwise, the tube flare may be completely cut off. This occurs at about five times the specified wrench torque on a single tighten-

ing and at two or three times the specified wrench torque for a small number of tightenings on 52SO tubing, which is the tubing most commonly used in aircraft hydraulic systems.

Because hydraulic struts are usually hinged to the structure and moved while under load, the fluid must be conveyed to them through flexible tubing. The industry largely uses a design of flexible tubing constructed with a Neoprene inner tube, a double cotton braid, and a Neoprene outer tube, having a long male tube fitting on one end and a swivel nut on the other.

Care should be taken in hose installations to avoid torsion in the hose because it reduces the life of the hose and may cause the end fittings to loosen. Excessively sharp bend radii will also cause premature hose failure. The minimum bend radius for a reasonable service life is about six times the outside diameter of the tube. For longest possible service life a minimum bend radius of at least nine times the outside diameter of the tube is required. Because hose is made from braided tube, it has a tendency to increase in diameter and shorten when pressure is applied. A properly made hose installation should allow for a shrinkage of at least 3.5 per cent of the length of the hose.

CHAPTER XIII

TESTING AIRCRAFT HYDRAULIC UNITS

There are two types of testing performed on aircraft hydraulic systems, (1) production testing and (2) experimental testing. Experimental tests are determined individually for each unit or system to be tested and usually consist of a pressure leakage test, a pressure strength test, an operation test, and a life test in which the unit is operated a sufficient number of cycles to ensure that it will have a satisfactory service life in the airplane.

These tests are performed on the first part made to a new design to ensure that the design is satisfactory. Experimental tests are also performed on new units sent in for test by parts manufacturers unless the part has a record of satisfactory service under conditions similar to those in which it will operate in the application for which it is being considered.

Production testing takes place all during the process of manufacture, from rough casting to installation in the finished airplane. The reason for the installation test, of course, is to ensure that the airplane system will operate satisfactorily and will not leak. Because it is expensive to replace units after they have been installed in the airplane, it is good practice to test parts during manufacture and assembly in order to detect any faulty parts before they are installed on the airplane.

The decision whether to test at any particular stage in manufacture depends on whether the cost saved by detecting leaks early is equal to the cost of testing all the parts. A typical manufacturer's estimate is that if an average of 1 part out of 100 leaks, that is a sufficient percentage to justify testing each part. This percentage is based on the proposition that it costs 100 times as much to remove, repair, and replace one leaky valve after it has been installed in an airplane as it does to bench-test one valve during assembly.

Because the objective of testing during manufacture is to

remove any possibility of leakage in the final installation, it is desirable that each test should be to a slightly lower pressure than the test which precedes it. Thus there will be no possibility that in later tests owing to a slight gauge error or lack of accuracy on the part of the tester the pressure will be higher than that of an earlier test and thus possibly show up leaks which did not show up in the earlier tests.

Tests should be specified on the following drawings: hydraulic installations, hydraulic subinstallations where necessary, bench assemblies (panels, etc.), unit assemblies, castings (subjected to pressure), and hydraulic lines.

When installed in the airplane, each portion of the hydraulic system should be tested to its critical relief pressure or 1.25 times the system operating pressure, and each preceding test should be to a higher pressure in accordance with the following table:

TABLE III.—TEST PRESSURES

Drawing	Test Pressure (Use Whichever Is Higher)
a. Hydraulic installation except return lines	Critical relief pressure or 1.25 × system operating pressure
b. Hydraulic-installation return lines	0.125 × system operating pressure *
c. Hydraulic subinstallations and bench assemblies except return lines	1.10 × critical relief pressure or 1.35 × system operating pressure
d. Hydraulic-subinstallation and bench-assembly return lines	0.135 × system operating pressure *
e. Unit assemblies except return ports	1.20 × critical relief pressure or 1.50 × system operating pressure
f. Unit-assembly return ports	0.15 × system operating pressure
g. Castings except return ports	1.35 × critical relief pressure or 1.65 × system operating pressure
h. Castings—return ports	0.165 × system operating pressure
i. Hydraulic pressure lines (on lines drawings)	2.00 × critical relief pressure or 2.50 × system operating pressure
j. Hydraulic return lines (on lines drawings)	0.25 × system operating pressure

* If reservoir is included, note that its connecting lines must be plugged to prevent bursting reservoir.

Care should be taken, where several identical units are subjected to different critical relief pressures, to use the higher critical value.

Where the part is not connected into a pressure system (as, for example, in the case of manually operated brakes), "system operating pressure" is taken as the maximum pressure possible under any conditions except temperature expansion, and the "critical relief pressure" is taken as the maximum pressure possible under any conditions. System operating pressure should be taken as two-thirds the "applied," or "limit," load when that figure is available, unless the above gives higher pressures.

From the testing viewpoint, any hydraulic unit may be considered as an aggregation of gaskets, packing, dashpots, lapped fits, and other details that require testing. These tests should be covered thoroughly by the test notes on the drawing, if they are not covered by the requirements of some applicable specification. Production test notes on drawings need not provide for tests to check strength or design, for these are covered in experimental tests. To be complete, the notes should provide for tests of (1) each detail where leakage or malfunction is possible and (2) features affected by adjustment or installation procedure. The following details require test notes:

Gaskets	Leakage at test pressure
Packing	Leakage at test pressure and at static pressure, *i.e.*, 5 to 7 lb./sq. in.
Valve seats	Leakage at operating pressure and at static pressure if it affects operation
Lapped fits	Leakage at operating pressure
Valve openings	Flow at a pressure drop that must be specified on the drawing
Friction	Cups and other packings
Pressure	Relief pressures
Handle load	Where dependent largely on friction

Each of the preceding details can be tested by some combination of port and position. Wherever possible, this information should be given in tabular form. The following is an example of some of the values that should be tabulated where applicable:

Test port	Ports plugged	Position	Pressure, lb./sq. in.	Maximum leakage	Flow (minimum)

A reasonable group of leakage tolerances is as follows:

1. Maximum leakage[1]—1 drop in 48 hr. External leakage in cockpit or cabin or other leakage that would normally come in contact with personnel.

2. Maximum leakage[1]—2 drops in 12 hr. Struts, etc., for external leakage.

3. Maximum leakage[1]—1 drop per hour. Struts, etc., for internal leakage. Valves, etc., internal where critical (as in accumulator circuit).

4. Maximum leakage[1]—10 drops per hour. Valves, etc., internal where not critical.

Note: Where leakage tolerances are specified as so many drops in a given length of time, it is not absolutely necessary to observe

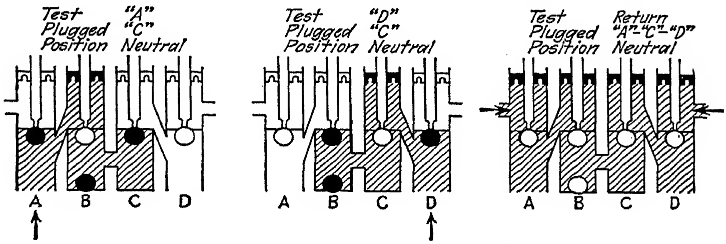


FIG. 91.—Steps in testing poppet four-way valve.

the leakage for that length of time. The tolerance may be interpreted normally as meaning the rate of leakage allowed. In other words, one-half the given quantity in one-half the given time, one-fourth the given quantity in one-fourth the given time, etc.

Figure 91 shows the three test steps necessary to pressure-test a poppet four-way valve, giving the details (filled in solid) that are tested under each of the three necessary test hookups. The test fluid is shown cross-sectioned. Note that a poppet four-way valve is (from the test point of view) simply an aggregation of cups and ball seats that must be tested.

It is good practice to test each unit twice and to operate once between tests. The second test should be for a period of at least 10 min., during which the unit should be inspected for leaks. If the leaks are around packings or gaskets, parts may be tightened

[1] Past packings and cups only. No leakage is allowed past gaskets, etc., or from porosity.

up to stop the leaks. If the leakage is through porosity in castings, it is necessary to disassemble the unit and treat it with sodium silicate under pressure. Any packing sealing external leakage must also be tested at a low pressure, approximately 5 lb./sq. in., for packings will sometimes leak at this pressure that will not leak at high pressure.

After testing is completed, the unit should be drained, enough fluid being left in it to prevent any possibility of corrosion. It should then be plugged, preferably with metal or plastic plugs, to keep dirt out of the unit until it is ready for installation in the airplane.

Because all individual parts, including units, lines, flexible hose, etc., are tested individually, the installation test on the airplane is for the purpose only of checking for leaks at tube fittings and other joints that are assembled on installation. For this test, it is desirable to hold the entire system under high pressure a sufficiently long time so as to allow it to be inspected for leakage at joints.

CHAPTER XIV

MAINTENANCE OF AIRCRAFT HYDRAULIC SYSTEMS

Maintenance of aircraft hydraulic systems is carried out by most airlines and by the Army and Navy by the "periodic overhaul" system. Certain periods of time, measured in hours of flying time of the airplane, are set up at which various hydraulic units are removed from the airplane, disassembled, completely inspected, and assembled ready for reinstallation. When one unit is removed, a new unit from stock is put in the airplane; and, after overhaul, the overhauled unit is replaced in stock. When this system is used, provided that proper overhaul periods are established, failures in operation are extremely rare.

Engine pumps and similar continuously operating units are usually replaced when the engines of the airplane are changed, which occurs on most airlines at approximately 600 hr. of flying time. Flexible hoses in the engine section, which deteriorate rapidly in the presence of heat and oil, are usually replaced at each engine change. Operating cylinders and their attaching hoses are replaced at periods varying from 1500 to 4500 hr.

As an example of typical overhaul periods, retracting struts on the Douglas DC-3 are changed by most airlines at 1500 hr.; flap struts, which are better protected and are not subjected to such severe service, are changed at 3500-hr. periods. Directional control valves, selector valves, relief valves, accumulators, and similar units are usually changed at periods of 4000 to 4500 hr. Lines and fittings and structural parts of the hydraulic system are normally changed only at time of overhaul of the complete airplane, which for the Douglas DC-3 has been set at 8000 hr. flying time. Of course, all component parts of the airplane are inspected at regular intervals. Usually the most frequent inspection interval is about 20 hr., when hydraulic units are only looked at to be sure that no externally visible failure has occurred. At 50-hr. intervals, parts are looked at a little more thoroughly. At about 100-hr. intervals, the parts are inspected as closely as they can be in-

spected without removal from the airplane. Periodic inspection and periodic overhaul procedure have almost entirely eliminated hydraulic failure in airline airplanes.

When units are removed for overhaul, they are disassembled, carefully cleaned, and inspected. Metal parts showing evidence of wear are measured to ensure that the wear does not exceed allowable limits specified by the manufacturer of the part. If the wear exceeds these limits, the part is either replaced or repaired by

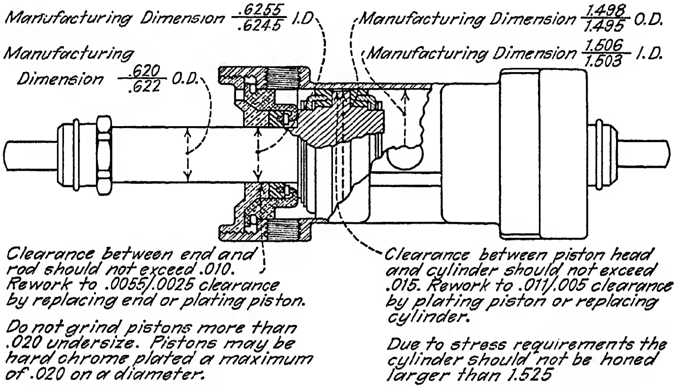


FIG. 92.—Maintenance drawing of cowl-flap operating cylinder.

chrome plating, metalizing, or some similar process for adding metal. Figures 92 to 94 show typical hydraulic-system maintenance drawings giving allowable wear limits. Figure 92 shows a typical operating cylinder, in this case the cowl-flap operating cylinder on the Douglas DC-3. This drawing shows manufacturing dimensions, gives clearances that may not be exceeded, and suggests one method of rework.

As an example, take the portion of the piston head between the piston packings. Manufacturing dimensions for the cylinder are 1.503 to 1.506. The manufacturing dimensions for the piston are 1.495 to 1.498; the maximum clearance is therefore the difference between 1.495 and 1.506, or 11 thousandths of an inch, whereas the minimum clearance is 5 thousandths of an inch. The maintenance drawing specifies that the clearance between the piston head and the cylinder should not exceed 15 thousandths, thus allowing 4 thousandths wear if the part was manufactured to the

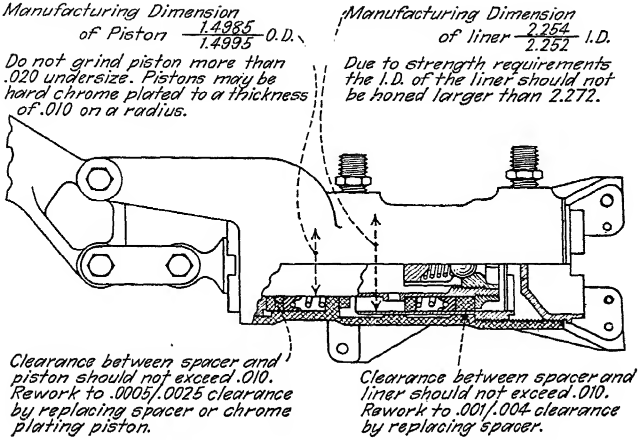


FIG. 93.—Maintenance drawing of hydraulic hand pump.

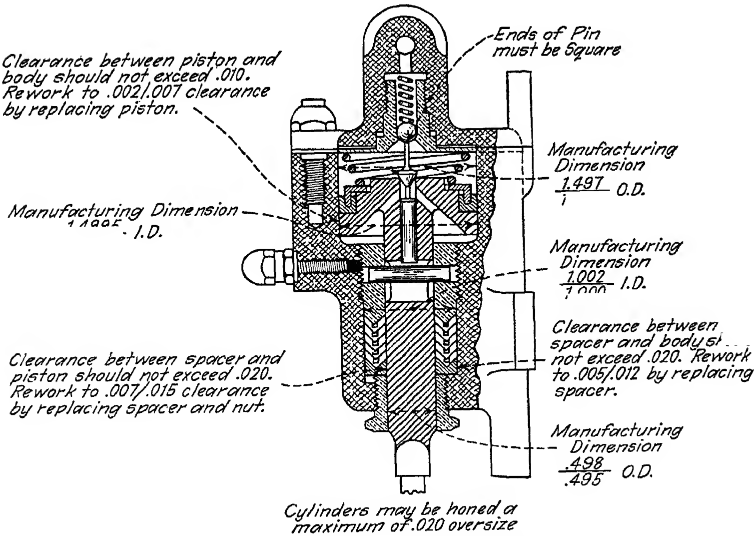


FIG. 94.—Maintenance drawing of power-brake valve.

widest clearance or 10 thousandths wear if the part was manufactured to the closest clearance. The manufacturing drawing also specifies that the piston may be plated or the cylinder may be replaced to bring the clearance back to within the manufacturing tolerance of 5 to 11 thousandths clearance. Similar instructions are given covering the clearances behind the rod packing.

Figure 93 shows the hydraulic hand pump for the Douglas DC-3 and gives manufacturing dimensions, clearances, and methods of rework for parts adjacent to the rod and the piston head packing cups. Figure 94, which shows the power-brake valve for the DC-3 airplane, gives manufacturing dimensions, clearances, and method of rework for the parts adjacent to the packings on the head and stem of the piston. In addition, other notes are given, such as that the end of the pin which lifts the pressure ball must be square. If allowed to wear off unevenly in service, these parts might cause irregular operation.

When the unit is reassembled after any worn metal parts have been replaced, all nonmetallic packings and gaskets are replaced and the unit is adjusted and tested in accordance with the procedure followed with a new unit. The unit is then stored ready to be installed in the next airplane coming in to have that particular unit overhauled.

In this volume, there have been discussed the theory of aircraft-hydraulic-system design, the laws of fluid flow, the design of aircraft hydraulic systems and of aircraft hydraulic units, and the drafting, manufacturing, installation, testing, and maintenance of hydraulic systems. Anyone desiring to pursue the study of aircraft hydraulic systems further can at present do so best by actual experience with such systems.

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